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Marine Engineering

FEBRUARY, 1906.

LONGITUDINAL BENDING MOMENTS OF CERTAIN LAKE STEAMERS.*

BY W. I. BABCOCK.

In 1895 I designed and built at Chicago the steamer *Victory* for the bulk-freight trade on the great lakes. She was then the largest vessel that had ever been built in fresh water and the first one to reach 400 feet in over-all length, her keel being 380 feet, beam 48 feet, and depth 28 feet, with a displacement of about 8,400 tons (net) on 18 feet draft of water. On this draft she carried 5,200 gross tons of iron ore.

The *Victory* was also one of the earliest lake ships to be built on the channel system, the floors, frames, web frames, beams, ties, and stanchions all being of channel section. The upper deck was of steel complete, without wood covering. The main-deck beams were spaced uniformly 8 feet apart, making a beam at each end of a hatch and one beam in the center between hatches, the hatches through the upper deck being 8 feet fore-and-aft and 24-feet centers. No deck was laid on the main-deck beams throughout the cargo hold, but a web frame or belt was fitted at each beam end and a stanchion on the center line, with two rows of ties on each side and steel-plate shifting boards below the upper deck on the stanchions. Large brackets were fitted at beam ends and center line.

Throughout the ship, extending from forward collision to after-peak bulkheads, was fitted a double bottom 4½ feet deep, of 10-pound plate, stiffened by Z-bars, and covered with 3 inches of oak flooring, supported by continuous longitudinals on top of floors. This bottom was divided by center keelson and high-plate floors into eight watertight compartments.

The machinery was placed as far aft as possible, consisting of a triple-expansion engine 23, 38, and 63 inches by 40 inches, and two Scotch boilers 14 feet 8 inches by 13 feet placed on a half-deck about 9 feet above the inner bottom, facing each other, with fire room between and coal bunker next forward. Accommodations for the crew and three or four passengers are all in forward and after ends of the ship, leaving the deck unobstructed as far as possible for convenience in handling cargo, which is all handled from the docks, there being no hoisting appliances whatever on the vessel.

THE STEAMER ELBERT H. GARY, OF THE UNITED STATES STEEL CORPORATION FLEET.

Thus built, the *Victory* ran for ten seasons, a successful and fortunate ship and a profitable one for her owners, mostly in the business of bringing iron ore from Lake Superior to Lake Erie ports and going back light, but occasionally, when freights were attractive, taking coal up or a load of grain down. In that time, however, and particularly in the last two or three years, there has been a great change in the conditions of lake business. This has been caused mainly by the great development that has taken place in the dock appliances for the unloading of bulk cargoes of iron ore. When the *Victory* was built, and for years afterward,

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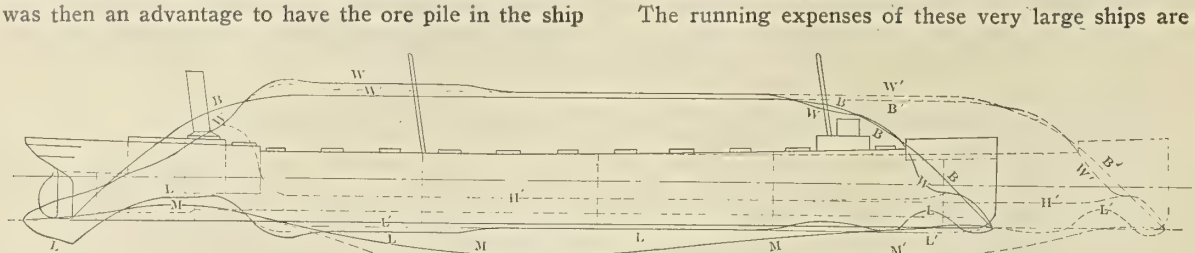
the unloading buckets used were ordinary tubs, holding about one ton, with four small wheels on the bottom, so that they could be moved about in the hold, and the ore was shoveled into them by hand. It was then an advantage to have the ore pile in the ship



CURVES FOR THE VICTORY IN STILL WATER.

as shallow as possible, so that the shovelers could get quickly to the bottom. Now, great hydraulically- or electrically-operated self-filling buckets or grabs, of five- to ten-tons' capacity, dig their own way into the cargo, and nearly all of it is unloaded without any hand work at all, the shovelers being required only at the last in cleaning up the scattered remnants. In operating this modern power bucket in the hold it is evident that the fewer obstructions there are the better, and that the hatches should be as large and as close together as possible. Also, that the ore pile should be concentrated on a small bottom area and, therefore, be deep or high. From these considerations there has been evolved the modern vessel, in which the hatches are 9 feet wide, spaced 12 feet centers, the main-deck beams and the stanchions omitted entirely, and the tank top bent up to meet the main-deck stringer in various forms, but all with the idea of forming a long hopper or trough in which the ore lies and which presents a smooth surface everywhere to the buckets. The amount of water ballast that can be carried is largely increased at the same time, which is a great advantage, as the boats usually go light to the upper lakes. The strength is made up by carrying a heavy plate girder or arch across under the upper deck between each two hatches, connected to heavy plate webs running right around the ship.

With the increase in the amount of cargo that can be unloaded in a day by this improved machinery and construction, and the steady increase in the draft of water available in the Soo and Detroit rivers and the lake harbors, due to government and private dredging, the size of the vessels has grown steadily also until now there are many on the lakes with a capacity of over 10,000 gross tons per trip each.

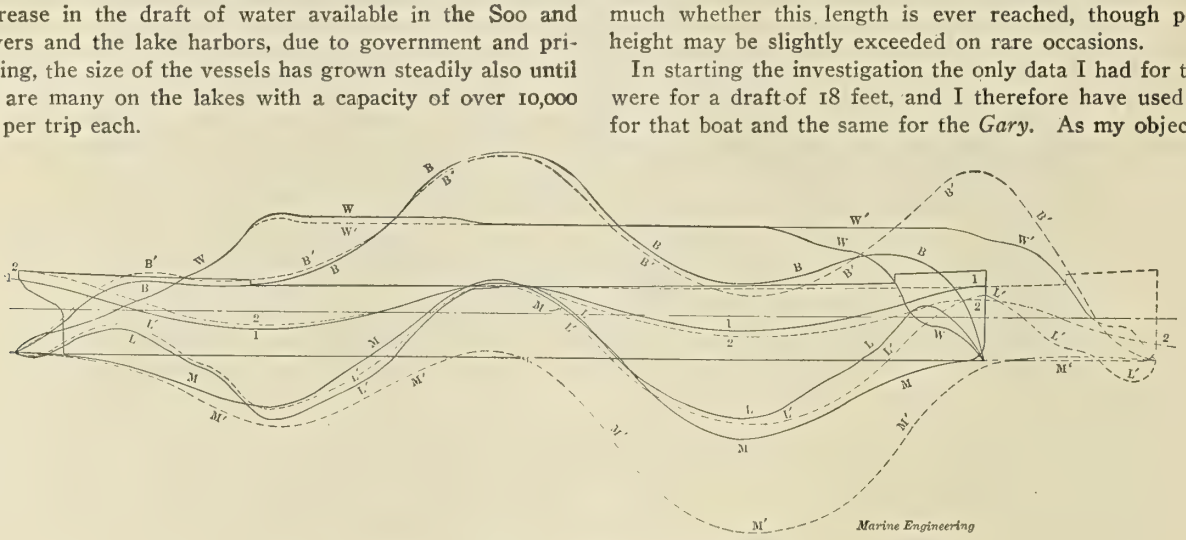
four hatches, fitted with telescopic covers made of steel plate and opened and closed by three special engines on deck through shafting, clutches, and chains.

The running expenses of these very large ships are very little more than those of the old boats, practically the same crew being carried, and it is evident, therefore, that the cost of carrying a ton of cargo is very much less. To bring down this cost in the old boats a number of them have been lengthened, increasing the capacity considerably with no increase in the running expenses and very little loss of speed. This was done last winter in the case of the *Victory*, 72 feet being added amidships, making her now 452 feet keel and giving an increase in cargo of some 1,400 tons on 18 feet draft. At the same time a new steel tank top was put in without wood covering and the main-deck beams were moved away from the line of the hatches, so as to leave as much clear space as possible for the operation of the unloading machinery.

It occurred to me that it would be interesting to investigate the changes in the longitudinal bending moments in a seaway in this ship caused by this increase in length, and to compare these also with the bending moments of the largest ships now in lake service for the same series of waves, and I now have the pleasure of presenting the results of this investigation to the society.

The waves met with on the great lakes of America are uniformly short, though they may be relatively high. From all the information I have been able to get, I fixed upon a wave 200 feet long and 20 feet high as the maximum size likely to be encountered, and made the calculations on that wave as a basis. I doubt very much whether this length is ever reached, though possibly the height may be slightly exceeded on rare occasions.

In starting the investigation the only data I had for the *Victory* were for a draft of 18 feet, and I therefore have used that draft for that boat and the same for the *Gary*. As my object was only



VICTORY ON WAVES WITH CREST AT ORIGINAL BOW.

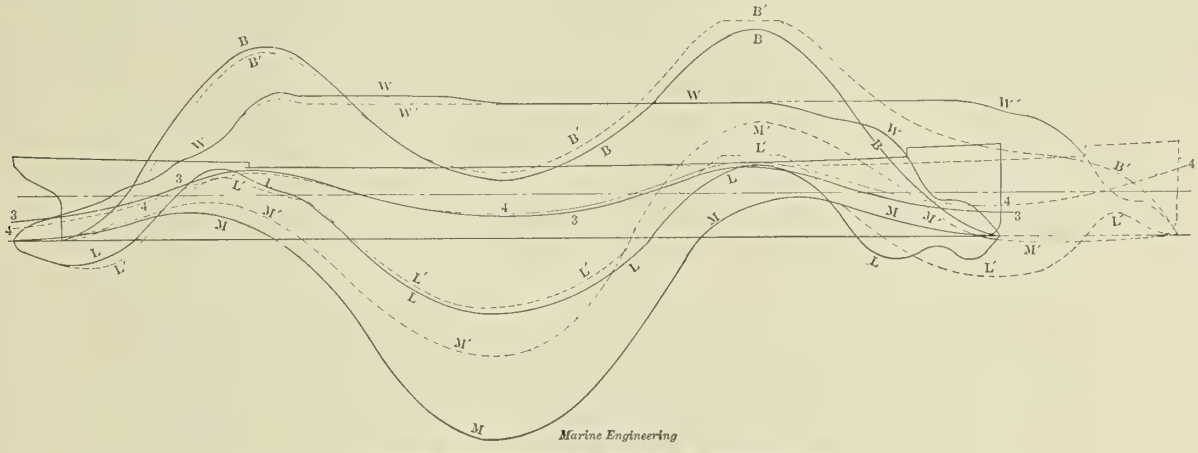
As an example of the latest boats, the steamer *Elbert H. Gary*, of the fleet of the United States Steel Corporation, may be taken. This ship is 569 feet over all, 549 feet keel, 56 feet beam, and 31 feet deep. She has a triple-expansion engine 24, 39, and 65 inches by 42 inches, and two Scotch boilers 15 feet 4½ inches diameter by 11 feet 6 inches long, and with 10,500 tons cargo on a draft of 19½ feet makes 12 miles per hour with about 2,000 horsepower. The cargo has been loaded in four hours and thirty minutes and taken out in four hours and ten minutes. This boat has thirty-

to determine the relative strains due to increase of length, and not the actual stress in the material, the actual draft for which the calculations are made does not make any particular difference. During the past season the boats have been loading to between 19 and 20 feet.

For each case the distribution of the cargo fore and aft in making up the curve of weights has been taken from actual records kept on the ship.

For the *Victory*, each figure gives an outline of the ship, dotted

for the increased length, the curves of weight, W and W', the water or wave lines, the curves of buoyancy, B and B', the curves of loads, L and L', and the curves of bending moments, M and M', for the original and the new lengths respectively, the position of the crest of the wave with reference to the stern being the same. For the *Gary* there is, of course, only one set of curves for each wave position.



VICTORY ON WAVES WITH HOLLOW AT ORIGINAL BOW.

These curves are given for the ship in still water, and in it I have added the weight curves for the vessel alone with 100 tons fuel, but without cargo. It shows, what is practically universal in lake ships, a sagging moment for the greater part of the length, when the vessel is loaded, which at its maximum, however, is only 7,000 foot tons for the original and 11,000 for the new length, being respectively 1-456 and 1-424 the product of the length and displacement.

The curves are also given for the ship in a series of waves with the crest of the first wave coming at the bow of the old length. The maximum hogging moment is 15,800 foot tons for the old and 1,000 foot tons for the new length, or 1-200 and 1-4,670 the product of the length and displacement respectively. The maximum sagging moments are 17,200 foot tons for the old and 37,100 foot tons for the new lengths respectively, or in fractions as before 1-186 and 1-126.

The ship is shown on a series of waves with the hollow of the first wave at the bow of the old length. The maximum hogging moment is 8,100 foot tons for the old and 23,200 foot tons for the new length, or 1-400 and 1-200 the product of the length and displacement respectively. The maximum sagging moments are 41,400 foot tons for the old and 24,500 foot tons for the new lengths respectively, or in fractions as before 1-77 and 1-190.

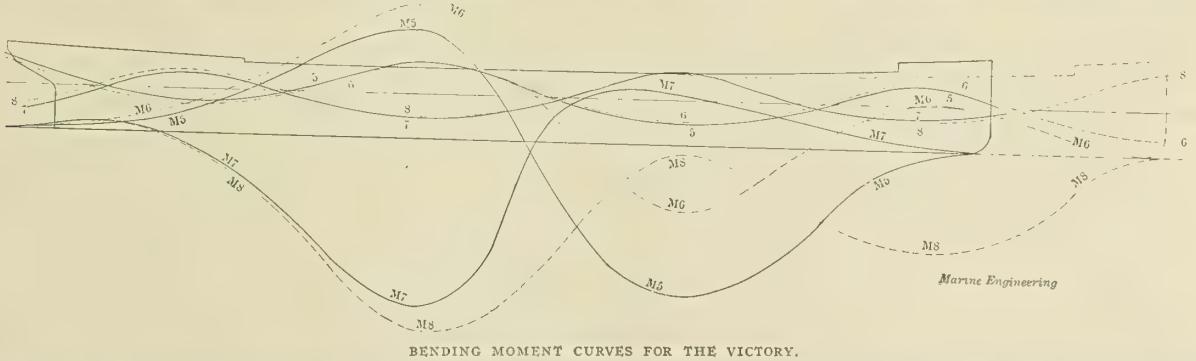
The bending moment curves only are shown in another plate, the others being omitted to avoid complication, for wave positions in which the crest and the hollow of the first wave come at the bow of the new length respectively. For the original ship the maximum hogging moment is 21,800 foot tons (1-152) and the maximum sagging moment is 34,700 foot tons (1-92). For the lengthened ship the maximum hogging moment is 26,700 foot tons (1-175) and the maximum sagging moment is 40,000 foot tons (1-117).

It is doubtless unnecessary to discuss these various figures in further detail. The drawings are plain and there are a sufficient number of different positions of waves illustrated to give the maximum bending moments which are ever likely to occur. From them we observe that the maximum hogging moment is about 21,800 foot tons for the original ship and 26,700 foot tons for the lengthened ship, an increase of about 22½ percent. The maximum sagging moment is 41,400 foot tons for the original and 40,000 foot tons for the lengthened ship. In any position which this vessel is ever likely to assume among waves it is fair to believe that the sagging moment will always be considerably in excess of the hogging moment.

For purpose of comparison with the curves of the *Victory*, similar calculations have been made on the steamer *Elbert H. Gary*, which is one of four sister ships which are now the largest in service on the lakes. Curves are given for this ship, lettered the same as in the case of the *Victory*. They explain themselves. The maximum sagging moment for the loaded ship in still water is 40,700 foot tons or 1-200 the product of the length and displacement. For the various waves shown the maximum hogging moment is 40,200 foot tons (1-200) and the maximum sagging moment is 97,500 foot tons (1-84).

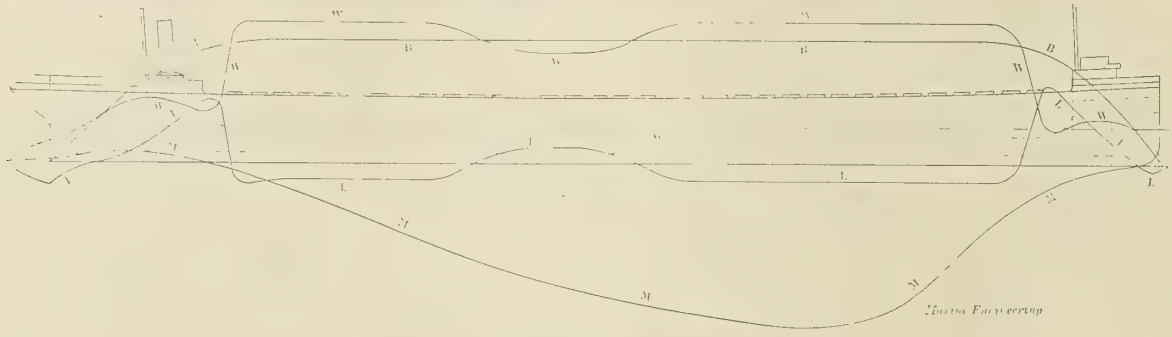
Comparison between the still-water conditions shows at once an important difference in the two ships. In the case of the *Victory* the forward hatch is considerably nearer the bow than in the *Gary*, and there being a cargo space underneath the boilers and coal bunkers, the ore from the after hatch trims aft underneath the boiler deck. This results in a more even distribution of the curve of weights for the loaded condition of the ship.

In the case of the *Gary* the boiler room and coal bunker extend down to the tank top and the ore at the after end can go very little aft of the line of the after hatch. At the forward end of the *Gary* the forward hatch is kept back from the forecastle



BENDING MOMENT CURVES FOR THE VICTORY.

house to give room for the winches which handle the wire hawsers. The result is that in this ship there is a considerable excess buoyancy both forward and aft of the cargo, whose result is seen in the curves which show a large increase in the sagging moment. This effect is aggravated by the practice on the *Gary* of not loading any ore at all into the forward hatch, which is done to facilitate the discharge of cargo, the unloading buckets, working through the forward hatch, being able to reach all of the ore, which would hardly be the case if any were loaded into that hatch. It is usual on the *Gary*, in order to reduce the sagging moment, to lighten up the amount of cargo loaded into the midship hatches. This is good practice, but leaving the cargo out of the forward hatch is not to be commended, though it is frequently done in lake ships.

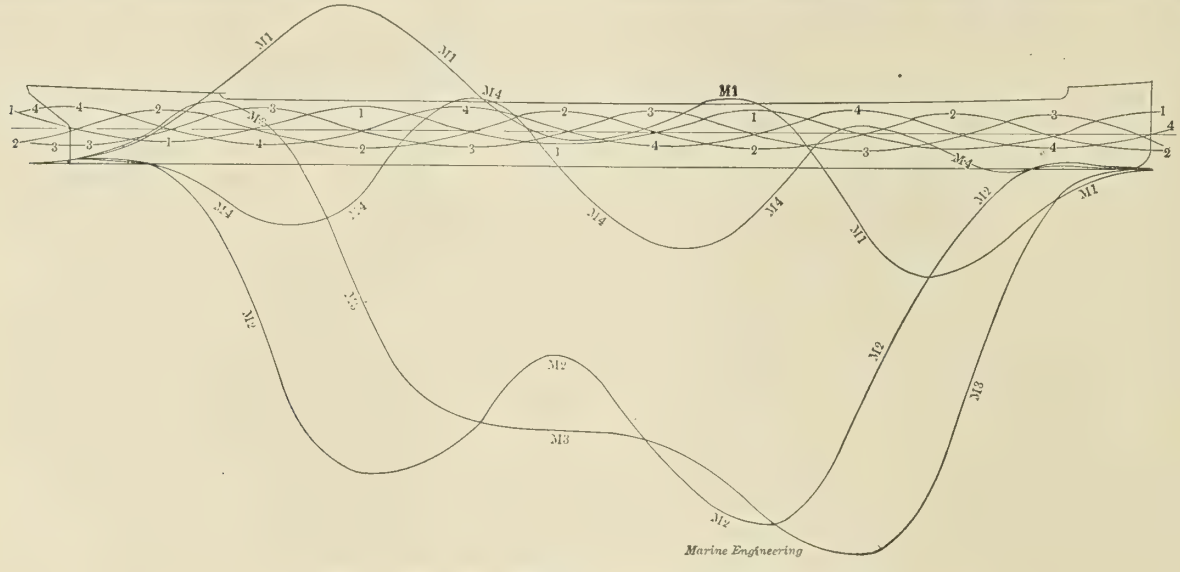


CURVES FOR THE ELBERT H. GARY IN STILL WATER.

In one case a crest of wave No. 2 comes nearly opposite the boilers, where there is already an excess of buoyancy in still water, as shown, and the result is seen in the increase of the sagging moment a short distance forward.

Similarly wave No. 3, drawn so that its crest comes just forward of the forward hatch where there is again a large excess of buoyancy in still water, causes another very large increase in the sagging moment a few hatches further aft.

A comparison of the various curves shown in the figures indicates that in any condition of the loaded vessel, in still water or among waves, the sagging moment is in excess, and generally very much in excess, of the hogging moment. Considerable criticism, therefore, which has at various times been indulged in as to the heavy bottoms of lake ships in comparison with their upper works, is shown to be largely unwarranted. This construction of the vessel has been a natural outgrowth of the conditions of the business, which require a very heavy bottom to resist the strains due to frequent grounding and touching in narrow and shallow channels, and also to support the weight of the cargo on the tank top, and also require the upper-deck hatches to be as numerous



THE GARY IN WAVES WITH CRESTS AND HOLLOWS RESPECTIVELY AT BOW

The conditions under which the iron-ore business of the lakes is carried on make it impossible to avoid these large excesses of buoyancy forward and aft of the cargo. In a service where the boats can run but seven months in the year despatch in handling cargoes is of the utmost importance. At upper-lake ports the ore is simply spouted in from elevated docks. Practically, therefore, everything about the boats is subordinated to the requirements of the unloading machinery at the receiving docks on the lower lakes. If a ship could be built so that she came to these docks with all her ore in one deep pile amidships and the whole upper deck could be taken off to leave this pile open to the dock machinery, possibly then the dock managers would be satisfied. Failing this ideal condition the shipbuilder has to come as near to it as possible and take care of the other qualities of his ship as best he can. I will venture to say, however, that nowhere else in the world is there a fleet of ships better fitted to the conditions under which they must work than on the great lakes, where boldness and originality of design have had freer scope or been more successful, or where, on the whole, fewer mistakes and failures have been made.

and as wide as possible to facilitate unloading. These combine to make the bottom heavy and the deck light, and therefore considerably lower the position of the neutral axis. It is satisfactory to know that the strains which come upon the ship in deep water among waves are very well met by the same construction, for the compression on the upper deck is always much more severe than the tension and to resist this compression it is comparatively easy to stiffen the stringer by one or more lines of intercostals between the beams, as shown in the midship sections illustrated herewith.

If the *Gary* were in service in salt water instead of on the lakes,

and therefore liable to encounter ocean waves which are, of course, much longer from crest to crest, the bending moments would naturally be greatly increased. For comparison I have calculated these moments for a wave of the same height as used before, 20 feet, and a length equal to the over-all length of the ship, 570 feet, with the crest and hollow at mid-length respectively, and the results are shown.

For the crest at mid-length the hogging moment is 165,300 foot tons or 1-33 the product of the length and displacement. For the hollow at mid-length the sagging moment is 257,900 foot tons or in fractions as before 1-21. The hogging moment is about the same as given by White as the maximum for ordinary merchant steamers. The sagging moment is between two and three times as great as given by the same authority. This is the natural effect of the full model of lake vessels—the box coefficient of the *Gary* at 18 feet being .86—and the excesses of buoyancy forward and aft as referred to before.

The greatest tensile strain on the deck of the *Gary*, therefore, is only one-fourth of what it would be if the ship were on salt water and in a series of waves of her own length. The maximum compression on the deck is a little less than two-fifths of what it would be under similar circumstances. If it were not for these facts the very wide hatches of lake steamers would be impossible and the handling of bulk cargoes would be greatly interfered with.

No attempt has been made to investigate the strains on either of these vessels when going up without cargo. From the nature of the case it is practically impossible to do so. In smooth water water ballast is carried aft only and added forward as the sea rises. There is no question but that in the past various vessels have been greatly abused, water ballast being carried aft only to get the wheel down and the vessel forced against head seas, jumping many feet of her bow clear of the water with a resultant heavy tensile strain on the deck. It is doubtless true also that many vessels are severely strained in unloading, too much ore being taken out at one place before the machines are moved. Definite calculations, however, cannot be made on such conditions.

Since the new power unloaders have brought the hopper type of vessel into existence the amount of water ballast that can be carried has been largely increased until on a vessel like the *Gary*

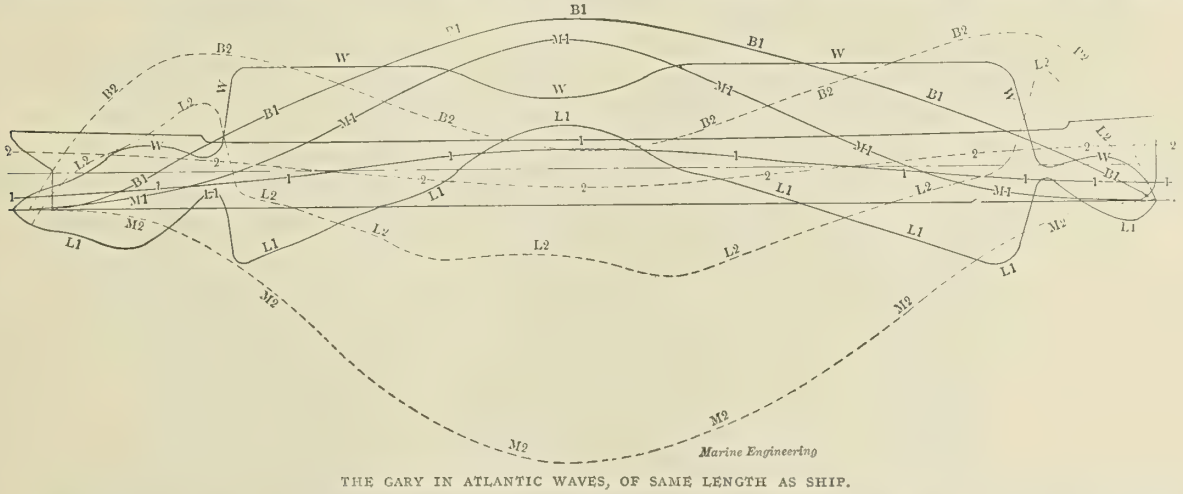
THE GARY, WHICH RECENTLY CARRIED A CARGO OF 12,320 TONS OF IRON ORE.

it amounts to about 8,000 tons and puts her down to a draft which makes the ship perfectly safe under all conditions.

The United States Steel Corporation has just placed an order for two ships 600 feet in over-all length, 58 feet beam, and 32 feet deep, to come out next year. To reduce as much as possible the excess buoyancy forward in these vessels, the forward hatch has been moved close to the forecastle and the third hatch omitted, the winches being placed on the deck in the space which that hatch would otherwise occupy, thereby carrying the ore body further toward the bow.

Editor's Note: The midship section of the steamship *Victory* has been taken as a basis for calculating the stress experienced in various parts of the structure of the ship due to her being placed upon the waves in question. It must be understood that such a calculation as this can give only approximate results, because no allowance can be made for either the corkscrew twisting motion to which ships are subjected when crossing waves diagonally, or the dynamic effects due to the buffeting of waves, from whatever quarter they may approach.

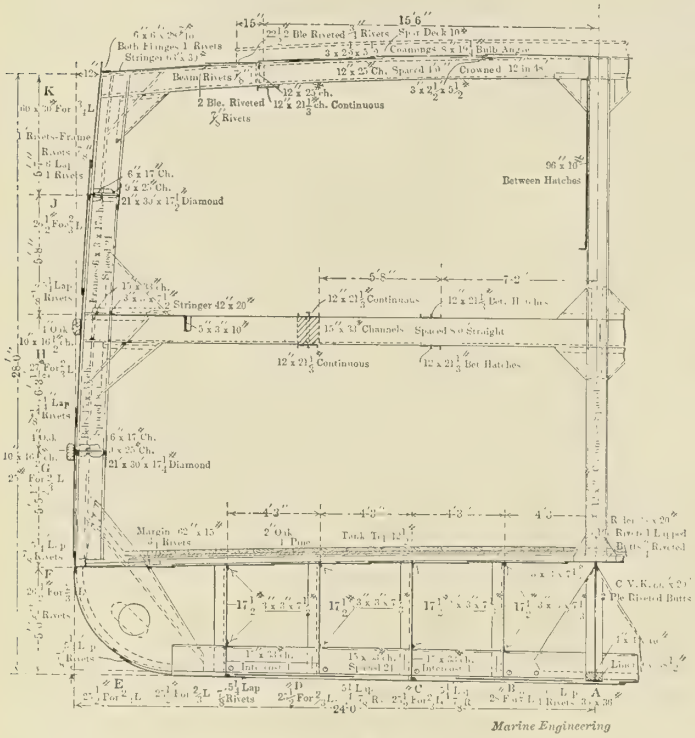
The data given on the midship section as illustrated in Mr. Babcock's paper was used for this computation, and trial neutral axes were taken at 10 feet 5½ inches above the base line for



THE GARY IN ATLANTIC WAVES, OF SAME LENGTH AS SHIP.

sagging and 8 feet 3 inches above the base line for hogging. The riveting, wherever it was not given in detail, was figured from the rules of the Great Lakes Register, and proper allowance made for rivet holes in all members subject to tension, these members being, of course, those above the neutral axis when the ship is undergoing hogging stress, and those below the neutral axis when the stress is of the sagging character.

As computed, the cross sectional area of all continuous longitudinal members is 1,532 square inches. The total rivet area is 329 square inches, and the net area after removing all rivet holes is 1,203 square inches. For sagging, the area figures out at 1,315 square inches; while the difference between the moments above and below the neutral axis shows the latter was assumed about 1¼ inches too high. Correcting for this difference, the total moment of inertia of the section under sagging conditions becomes 133,974, in the conventional units made up of square inches times square feet. For hogging, the net area of the section is 1,468 square inches, and the correction of neutral axis due to incorrect assumption is less than ¼ inch. The resultant moment of inertia of the entire section for hogging becomes 122,434 units, as above described.



MIDSHIP SECTION OF VICTORY.

A number of results are given quantitatively in Mr. Babcock's paper, being the maximum bending moment under each of several conditions. These quantitative results, however, do not cover completely all the cases which he investigated, and the others included in the table herewith have been deduced from his curves as reproduced. From the same curves have been obtained the positions with regard to the midship section of the ship of the points at which the maximum bending moment occurs in each instance. It will be noticed that these distances vary markedly, in some cases being in the neighborhood of 100 feet forward or aft of the midship section. With the long dead-flat sections of lake steamers, it is a fair assumption that the midship moment of inertia will be maintained for a long distance forward or aft of amidships; and even after the area of section begins to undergo change the moment of inertia of the new form of section will not vary much from that amidships. On this assumption, the stresses have been calculated for both keel and upper deck stringer, for sagging and hogging conditions of the ship, and with five different wave conditions for both the original ship and the ship finally lengthened. These conditions are respectively: still water; wave crest at bow of original ship; at bow of new ship; wave hollow at bow of original ship; and at bow of new ship. The distances from the corrected positions of the neutral axes to the keel are; for sagging, 10.47 feet; for hogging, 8.39 feet. For the upper deck stringer these distances are respectively 15.76 and 17.89 feet.

By use of the usual formula, $p = \frac{Me}{I}$, where p is the stress in tons per square inch, M is the bending moment in foot tons, e is the distance in feet from the neutral axis to the fiber under investigation, and I is the moment of inertia of the section in units as given above, the stresses above mentioned have been calculated, and are shown in the table. It will be noticed that the highest stress given for the original ship is 4.87 tons per square inch, which occurs in the deck stringer under sagging conditions with the wave hollow at the original bow. The highest value for the lengthened ship is 4.7 tons per square inch, which occurs under sagging conditions with the crest of the wave at the new bow. Both these values are very moderate as compared with those experienced in similar structures on the sea board, and of course all the other values, being considerably smaller in quantity than these, are such as need cause no uneasiness whatever regarding the strength of the ship.

RESULTS FOR ORIGINAL SHIP.

CALCULATED STRESS.

Position of wave.	Sagging Moment		Hogging Moment.		Sagging.		Hogging.	
	Maximum	Position.	Maximum	Position.	Keel.	Stringer.	Keel.	Stringer.
No wave.....	7,000	15 ft. aft	none	.....	0.54	0.83	.....	.....
Crest at original bow..	17,200	90 ft. ford	15,800	10 ft. aft	1.35	2.02	1 08	2.31
Hollow " " " ..	41,400	15 ft. aft	8,100	110 ft. ford	3.24	4.87	0.55	1.18
" " new " ..	30,100	70 ft. ford	21,800	45 ft. aft	2.36	3.55	1.49	3.17
Crest " " " ..	34,700	40 ft. aft	11'300	50 ft. ford	2.71	4.09	0.77	1.65

RESULTS FOR LENGTHENED SHIP.

No wave.....	11,000	55 ft aft	none	.....	0.86	1.29	.....	...
Crest at original bow..	37,100	55 ft. ford	1,000	45 ft. aft	2.9	4.36	0.07	0.15
Hollow " " " ..	24.5 10	50 ft. aft	23,200	45 ft. ford	1.92	2.88	1.59	3.38
" " new " ..	13,700	35 ft. ford	26,700	80 ft. aft	1.07	1.61	1.83	3.88
Crest " " " ..	40,000	70 ft. aft	1,900	near stern	3.	4.7	.....	.....

THE NEW P. & O. LINER DONGOLA.

BY BENJAMIN TAYLOR.

A new addition to the world-famous fleet of the Peninsular and Oriental Steam Navigation Company is always an event of the first importance and interest in the shipping world, because these steamers, carrying the whole of the mails and the best of the passenger and cargo traffic between Great Britain and the Far East, always embody the latest developments in marine architecture and equipment, and the most modern and approved achievements in marine engineering. The *Dongola*, which is the latest addition to this extensive fleet, was launched on the 14th of September, 1905, from the yard of Barclay, Curle & Company, Limited, Clydeholm, near Glasgow. This vessel is of a class specially designed for the Eastern mail service of the P. & O. Company, and has a combination of speed and spacious passenger accommodation, with large measurement and deadweight carrying capacity. She presents in one structure qualities formerly obtainable only in two or more different types.

The leading dimensions of the *Dongola* are:

Length, overall ..... 490 feet.
Breadth, extreme ..... 56 feet, 5 inches
Depth from bottom of keel to top of deck at center ..... 36 feet, 8 inches
Gross registered tonnage, about..... 8,100 tons.

DECK VIEW OF DONGOLA DURING CONSTRUCTION.

transport service have also been complied with, so that the vessel is available for the transport of troops in time of war.

The hull is of the best Siemens-Martin open hearth mild steel, the scantlings being in excess of the requirements of Lloyd's, with the object of providing increased strength and also of providing a margin against the deterioration common to steel vessels. The hull is divided into separate watertight compartments to insure the flotation of the vessel should the skin be accidentally pierced; and to provide for safety in the event of grounding or touching a sunken rock, a complete inner bottom has been fitted fore and aft, divided by numerous watertight partitions into separate water-ballast tanks, each tank being capable of being filled or emptied independently.

There are three decks laid fore and aft; a lower deck of pitch pine, a main deck of steel sheathed with pitch pine, and an upper deck of steel sheathed with teak. Above the upper deck is, at the fore end, a forecastle 103 feet long, and, at the after end, a long combined poop and bridge deck 320 feet long. Both of these decks are of steel, sheathed with teak. The combined poop and bridge forms the lower promenade deck, and above this is the upper promenade deck of teak, 190 feet long, assigned to

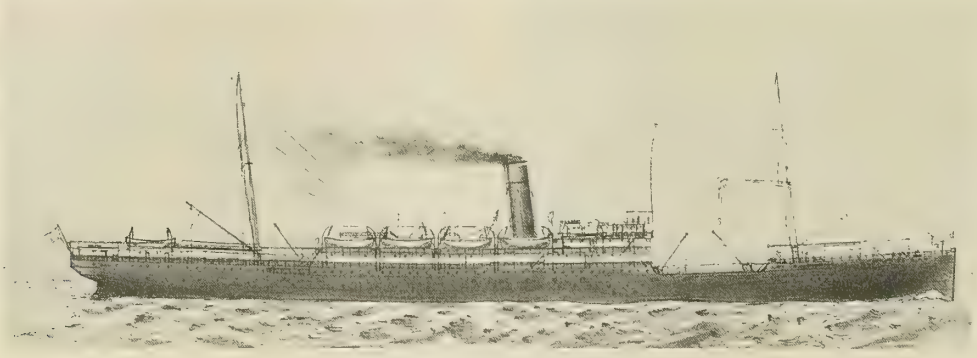
BOW VIEW OF NEW SHIP JUST BEFORE LAUNCHING.

The machinery consists of quadruple expansion twin-screw engines of 8,000 indicated horsepower, constructed by the builders, developing a speed at sea of 15½ knots. She will carry 150 first-class and 100 second-class passengers. The requirements of the first-class passengers. Above this are the boat deck, lower navigating deck, and upper navigating deck, all of teak. The total depth of the ship to the upper navigating deck is 80 feet, or 56 feet above the level of the water.

The arrangements for passenger accommodation are designed to give the maximum space and comfort to each individual. The first-class passengers' cabins are all amidships, partly on the upper deck and partly on the promenade deck. Only one row of cabins is fitted on each side amidships, instead of the usual practice of two rows each side. The passages are unusually wide, well lighted, and well ventilated. The dining saloon is situated at the fore end of the upper deck, and extends the full breadth of the ship. There are large circular lights on three sides of the saloon, and an open well overhead, which extends through four decks to a large skylight on the upper navigating deck. The decoration of the saloon is in good taste, the walls being paneled in inlaid mahogany, with specially designed Tynecastle canvas for the frieze and ceiling. The furniture consists of chairs, tables, settees, and sideboards of inlaid mahogany with beveled glass mirrors. Electric bells, fans, and lights are provided here as elsewhere throughout the vessel. Leading from the dining saloon to the music room on the upper promenade deck is a handsome stairway in two flights, with a large open space on the lower promenade deck, giving access to the deck cabins. The music room is large and decorated in inlaid mahogany, with Tynecastle canvas panels on the ceiling and frieze on the walls. It is well lighted by numerous lights on four sides and by two large skylights overhead. This room is furnished with piano, couches, tables, etc. Direct access is obtained from it to the upper promenade deck, and from the halfway landing to the lower promenade deck. At the after end of the upper promenade is the smoking room, which is lofty, the ceiling being two feet higher than any other room in the ship. The paneling is of light oak, and a deep frieze and ceiling of Tynecastle canvas fitted. Numerous couches, chairs and tables divide up the space into small compartments. Access to the smoke room is obtained through a vestibule lined in fumed oak, from which a stairway leads down to the after vestibule.

The sanitary accommodation is in numerous small blocks, each complete in itself, at different points throughout the passenger accommodation. Porcelain baths with hot and cold water, hot and cold showers, marble topped wash basins with hot and cold water, and all other necessaries, are provided in large numbers, and passengers can go from their rooms to any of the public rooms without going on deck.

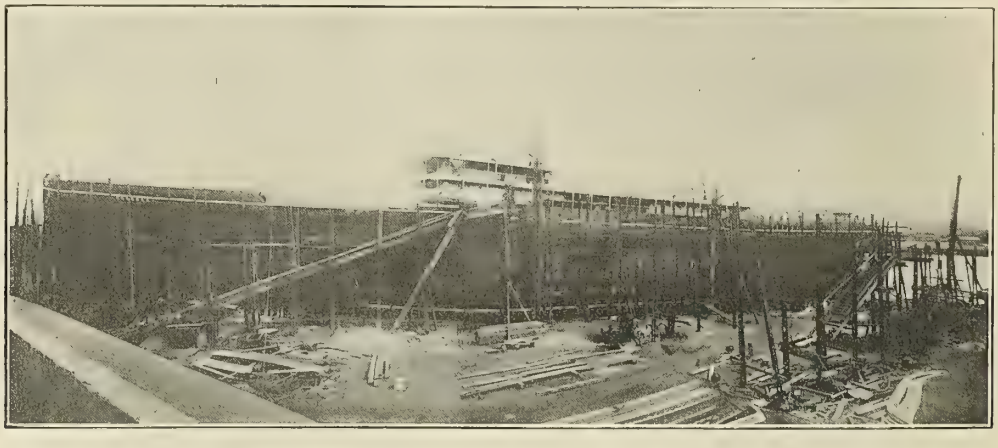
The second-class passengers are accommodated in the after end of the vessel, and the dining room, smoke room, entrance hall, and sanitary accommodation are all arranged with the same care and attention as the first-class.



THE DONGOLA, READY FOR SEA.

For the stewards' department large store rooms are provided on the lower deck, and a refrigerating chamber, with separate compartments for preserving meat, game, fish, and vegetables, also an ice-making room. The temperature of these chambers is kept at any desired point down to freezing point by means of a refrigerating machine on the dry-air system in the engine room. The galley is on the upper deck, and is fitted up with cooking ranges, baking ovens, hot water boilers, grills, *bain-maries,* etc. Large pantries are fitted adjoining each dining saloon, with hot and cold water, and carving tables heated with steam.

The electric light installation consists of three independent dynamos, each driven by a compound engine, and a complete system of electric lighting and fans throughout the vessel, including navigating lights and large clusters for use when discharging cargo at night. The ventilation of the vessel is designed for the hot climate in which she will run, and includes a special system of ventilation to the troop deck.



SIDE VIEW OF THE SHIP IN HER BUILDING BERTH.

The accommodation for the captain and officers is entirely on the navigating deck; a large chart room and wheel house contains the steering wheel and other navigating appliances, Kelvin compasses, engine room telegraphs, docking and steering telegraphs, speaking tubes to the engine room and captain's room, and loud-speaking telephones to the engine room, forecastle head, and the after end of the vessel complete the appliances for working the ship. The engineers are berthed on the upper

deck, alongside the entrance to the engine room, with their mess room and lavatories alongside. The seamen, firemen and stewards are all berthed forward, partly under the forecastle deck and partly under the upper deck.

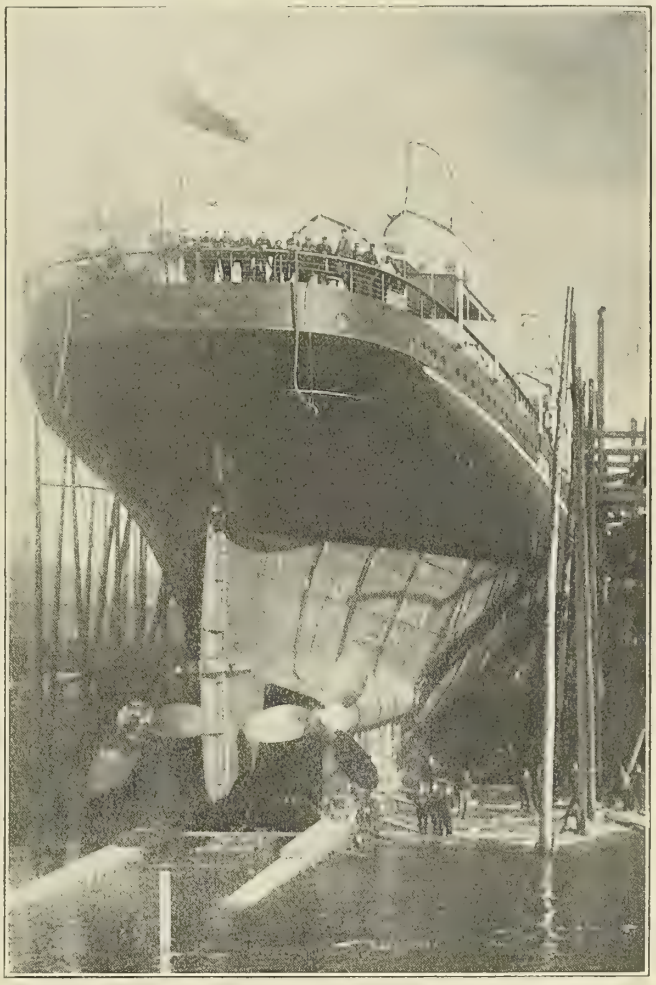
As the vessel is designed to serve as a troopship when required, the troop deck extends full length below the upper deck, two watertight doors being fitted in each watertight bulkhead to allow an uninterrupted passage from end to end, and numerous large side lights are fitted fore and aft. Ample latrine accommodation is placed at the fore end, under the forecastle.

The steam steering gear is fitted in a steel house immediately over the rudderhead, and is actuated by telemotor gear from the bridge and aft; and hand steering gear is fitted for use in the event of a breakdown of the steam gear. The appliances for mooring the ship consist of cast-steel stockless anchors, stowed in the hawse pipes, with heavy cables of the best quality; a large steam windlass on the forecastle head, with a steam capstan forward; and two steam warping capstans placed on the poop deck aft.

The cargo loading and discharging facilities are five large cargo hatchways served by six powerful steam cranes, four steam winches, powerful derricks, and a special derrick capable of lifting 30 tons.

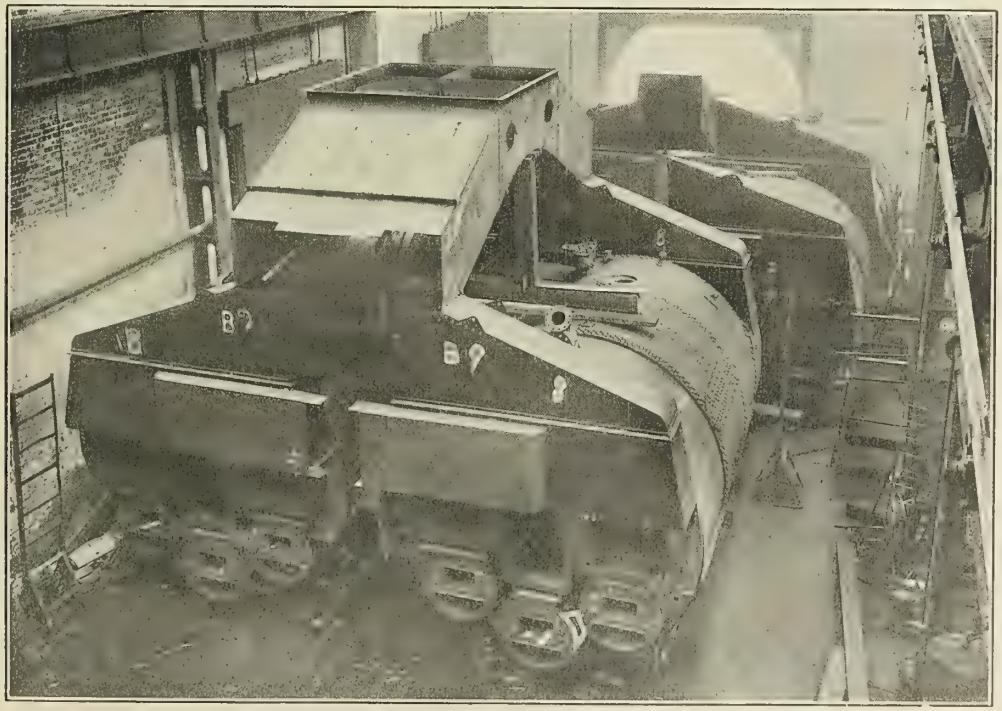
The ship carries 14 lifeboats, fitted with Andrews' patent lowering and disengaging gear, and those amidships are fitted on the rail of the promenade deck on Andrews' principle, so as to leave as much deck space as possible for the passengers. The boats include a large steam launch, capable of steaming 10 knots. Mail rooms and bullion room are fitted up on the lower deck aft, built of steel and fitted with Chubb's doors. The space between the twin screw tunnels is utilized for fresh water tanks, the total capacity being about 40,000 gallons.

The machinery, constructed by the builders, consists of two sets of quadruple expansion engines, the diameter of the cylinders being 25½, 36½, 52 and 74 inches, with a stroke of 51 inches. The cylinders are arranged to give the best balance possible for smooth and silent working. The second intermediate cylinder and the low-pressure cylinder are both steam jacketed; flat slide valves are fitted to the second intermediate and low-pressure cylinders, and piston valves to the others. Direct steam starting and reversing gear is fitted, also steam and hand gear for turning the engines in port. The surface condenser is horizontal, of cast iron, the condensing surface being 12,000 square feet.



STERN VIEW BEFORE LAUNCHING.

The shafting throughout is forged from the best ingot mild steel, about 10 percent in excess of Lloyd's requirements. The



THE BATTERIES OF BOILERS, SET UP IN THE SHOP.

propellers are three-bladed, with cast iron bosses and blades of manganese bronze.

The boilers are six in number, two double-ended 20 feet long, and four single-ended 11 feet 6 inches long; all 14 feet 8 inches diameter, with a working pressure of 215 pounds per square inch. Each boiler is fitted with three furnaces in each end, 3 feet 5¼ inches in diameter. Forced draught on Howden's system is fitted, with two fans electrically driven. The total number of furnaces is 24, the heating surface 18,700 square feet, and the grate surface 470 square feet; ratio, 39.8 to 1. The machinery is estimated to develop 8,000 indicated horsepower on trial, and a speed of 16 knots.

The outfit of auxiliary machinery in the engine room consists of fourteen steam pumps, including circulating and air pumps, feed, bilge, ballast, wash deck, fire, hot and cold fresh water and sanitary pumps. Steam ash ejectors, feed water evaporator, fresh water distilling, and evaporating and filtering plant are fitted in the engine room, also the refrigerating machinery and a triplicate set of electric generating plant.

The keel of the *Dongola* was laid on the 10th of January, 1905, and she was ready for launching by the 14th of July, but owing to a strike of patternmakers the machinery was not then ready. Even with this delay, the vessel was handed over to the owners ready for service within nine and a half months from the laying of the keel. But for the strike referred to, she would have been ready in eight months from beginning work.

Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, report under date of January 10, 1906, the following percentage of completion of vessels building for the United States Navy:

			Dec. 1.	Jan. 1.
BATTLESHIPS.				
Virginia	19 knots.	Newport News Co.	97.	98.
Nebraska	19 "	Moran Brothers Co.	83.	84.24
Georgia	19 "	Bath Iron Works	90.15	91.24
New Jersey	19 "	Fore River Shipbuilding Co.	93.7	95.5
Rhode Island	19 "	Fore River Shipbuilding Co.	98.	99.
Connecticut	18 "	Navy Yard, New York, N. Y.	92.66	93.69
Louisiana	18 "	Newport News Co.	92.56	94.17
Vermont	18 "	Fore River Shipbuilding Co.	68.7	70.7
Kansas	18 "	New York Shipbuilding Co.	67.7	69.8
Minnesota	18 "	Newport News Co.	77.1	80.33
Mississippi	17 "	Wm. Cramp & Sons.	45.74	47.32
Idaho	17 "	Wm. Cramp & Sons.	42.58	45.39
New Hampshire	18 "	New York Shipbuilding Co.	29.2	33.4
ARMORED CRUISERS.				
California	22 knots.	Union Iron Works.	86.2	87.5
South Dakota	22 "	Union Iron Works.	84.9	86.2
Tennessee	22 "	Wm. Cramp & Sons.	90.02	91.66
Washington	22 "	New York Shipbuilding Co.	89.4	91.9
North Carolina	22 "	Newport News Co.	26.77	30.74
Montana	22 "	Newport News Co.	22.29	25.84
SEMI-ARMORED CRUISERS.				
St. Louis	22 knots.	Neafie & Levy Co.	83.62	86.
Milwaukee	22 "	Union Iron Works.	86.	87.9
SCOUT CRUISERS.				
Chester	24 knots.	Bath Iron Works.	13.27	15.53
Birmingham	24 "	Fore River Shipbuilding Co.	15.4	18.2
Salem	24 "	Fore River Shipbuilding Co.	14.8	17.6
SUBMARINE TORPEDO BOATS.				
Submarine No. 9	.....	Fore River Shipbuilding Co.	43.5	46.9
" " 10	.....	Fore River Shipbuilding Co.	36.9	37.8
" " 11	.....	Fore River Shipbuilding Co.	37.	39.9
" " 12	.....	Fore River Shipbuilding Co.	33.6	38.7

Personal.

R. C. Monteagle, who has been connected for several years with the Gas Engine & Power Company and Chas. L. Seabury & Company, Consolidated, has resigned to become chief engineer of the Atlantic Works, East Boston, Mass.

Warren Johnson has resigned his position as manager of the Johnson Iron Works, New Orleans, and has opened offices as a consulting engineer. His address is P. O. Drawer 241, New Orleans, La. He is a graduate of Cornell University in the class of 1897.

THE DETERMINATION OF THE PRINCIPAL DIMENSIONS OF THE STEAM TURBINE, WITH SPECIAL REFERENCE TO MARINE WORK.*

BY E. M. SPEAKMAN.

In designing any turbine installation, the first and most essential step is to estimate the highest suitable speed of rotation in order that the turbine may be made as small as possible for any given efficiency. While there is now no great difficulty in accurately determining the proportions of turbines and propellers to insure certain results, with ample confidence to enable a stiff guarantee to be made both for speed and economy, the principal dimensions involve considerable calculation, and much attention must be devoted to the efficiency of propulsion if the best results are to be obtained. Knowledge of propulsive efficiency and all its component values for various classes of work is essential for calculating propeller dimensions, and thereby arriving at the highest revolutions attainable, for it is only when the rotary speed is settled that the best compromise of turbine dimensions, chiefly in the matter of blading arrangements, can be determined.

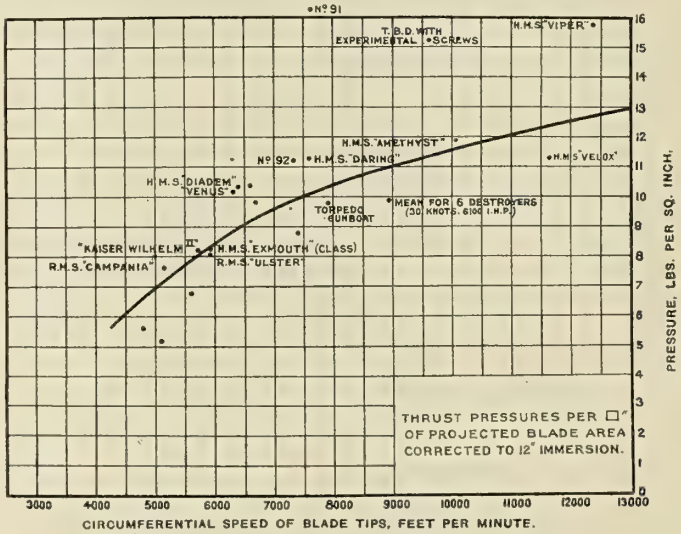


FIG. 1. PRESSURES ON PROPELLER BLADES.

Turbine efficiency and propeller efficiency must be considered separately and also together, because it may be found that the use of revolutions somewhat below the maximum obtainable will increase the combined efficiency, while on the other hand, to obtain certain advantages in weight and space, this efficiency may be slightly sacrificed at the highest speed, and it is necessary to know the effect of such modifications on the design and performance. Roughly, the weight of the turbines will vary inversely as the square of the revolutions, while the economy of the turbine will remain almost constant if designed for the same internal conditions; the efficiency of the propeller will be slightly improved as the revolutions decrease and the diameter is made greater.

In electrical work the design of generators almost invariably admits of very convenient speeds, but in marine work the size of propeller necessary to avoid cavitation imposes a much lower limit, with which the size of the turbine is greatly increased. The minimum size of propeller required to avoid this phenomenon (and its attendant inefficiency due to increased slip and consequent loss of thrust at the higher powers), must be calculated at the beginning of any design, as it is almost impossible to assume certain revolutions and later on to design a propeller to suit.

Cavitation is partly the result of attempting to obtain too much work per square foot of blade area, and partly of excessive

*A paper read October 24, 1905, before the Institution of Engineers and Shipbuilders in Scotland.

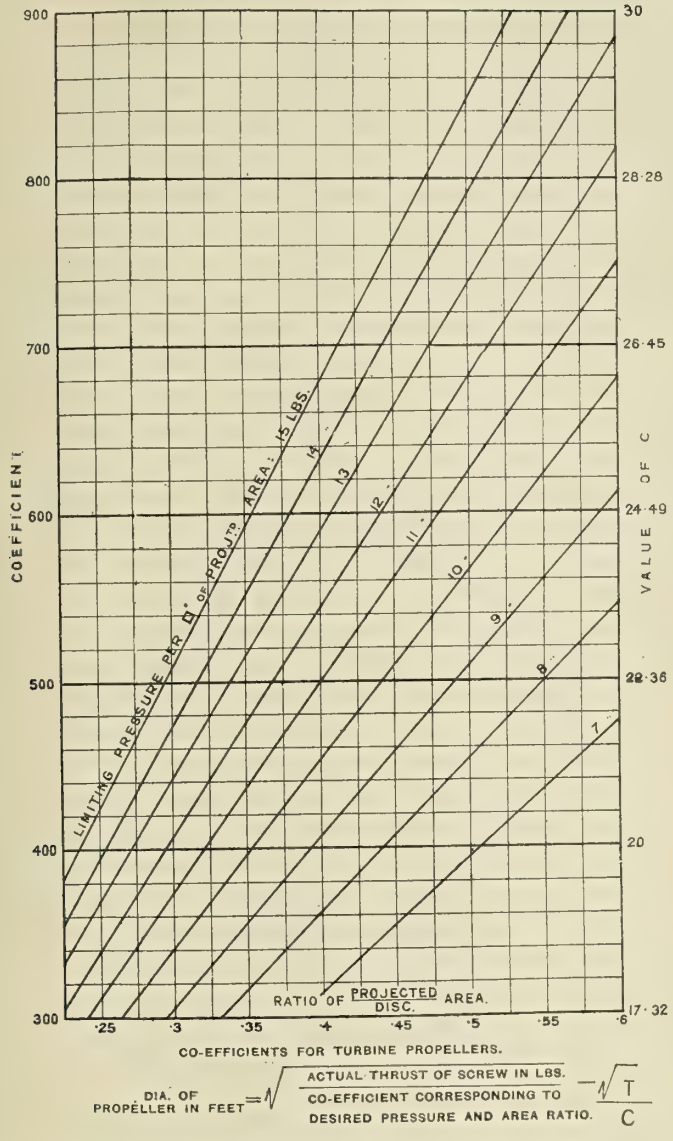


FIG. 2. TO DETERMINE TURBINE PROPELLER DIAMETERS.

peripheral speeds. It has been found, by bitter experience occasionally, that there is a narrow limit to the tensional pressure possible on the water, per unit of projected area, beyond which the propeller efficiency drops very rapidly. This pressure is approximately from 10 to 12 pounds per square inch at a depth of twelve inches below the surface, and to reduce the total thrust to this, sufficient blade area must be provided, which, in conjunction with certain practical proportions, necessitates a certain size of propeller, thereby limiting the revolutions.

Friction and slip constitute the normal losses in all propellers, and augmented resistance must also be taken into account. This latter loss, however, is materially reduced with the smaller diameters of propeller found in turbine work. Comparing the *Manxman* with the *Ulster*—a Holyhead mail boat of similar speed and power—the total disk area of the twin propellers of the latter is 226 square feet (for two 12-feet-diameter propellers), while that of the *Manxman* is only about 80 square feet. If the thrust deduction is proportional to the absolute area of the disturbance of the stream lines at the stern, the effect on the *Ulster* will be far greater than on the *Manxman,* but this action is to some extent affected by the intensity over the disturbing area, which again is modified by the proximity of the propellers to the side of the vessel, this being less in turbine work. The disk area in H.M.S. *Velox* is less than half that of the propellers in ordinary destroyers of the same power and speed. Cavitation is a preventable loss, and its presence on many vessels with insufficient blade area may be deduced from the falling off of the thrust curve and the rapid rise in the slip curve above a certain speed.

From the analysis of numerous trials it appears that the pressure per square inch of projected area, when reduced to twelve inches immersion of tip, due to the effective thrust, is approximately 1 pound for every 1,000 feet per minute of circumferential velocity of blade tips. For a screw of a given pitch ratio, working at its maximum efficiency, this velocity should be proportional to the designed speed of the ship, and at full speed the pressures seem to have hitherto been about 5 pounds per square inch for slow cargo vessels, from 6 to 7 pounds for ocean-going mail steamers, and from 7.5 to 8.5 for cross-Channel steamers; in cruisers and battleships they vary from 8 to 10.5 pounds in some recent notable instances,, and in torpedo craft from 9 to 11 pounds. At about 9 pounds—rather less, possibly—the lowest suitable limit is reached for turbine screws; from 10 to 11 pounds may be more usual in fast vessels, and though even from 12 to 14 pounds has been known, pressures over about 11 pounds always seem to be accompanied by low efficiencies. In large ocean-going vessels, which may be

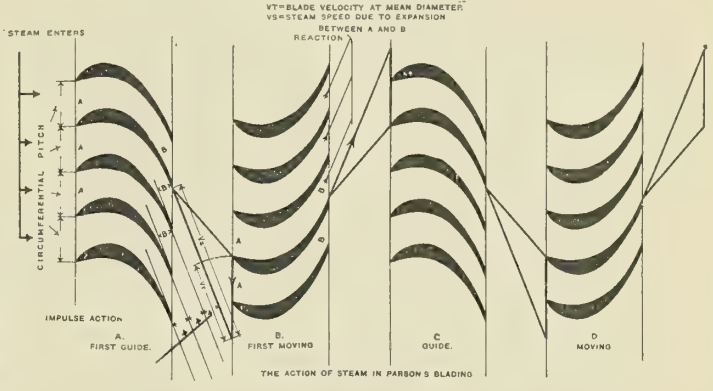


FIG. 4. IMPULSE AND GUIDE BLADE DIAGRAM.

delayed by headwinds and seas, a much lower designing pressure should be used, but in destroyers, as suggested above, something may be sacrificed at the maximum speed to obtain other advantages. The above pressure is worked out as mean pressure, as there exists no method of determining the local intensity per square inch, though the tendency of the distribution may be assumed in some cases. The only published results are in

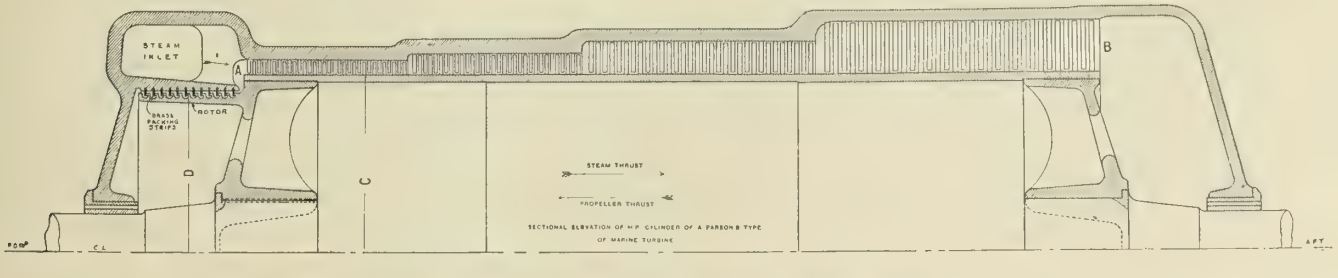


FIG. 3. SECTIONAL ELEVATION OF A PARSONS' MARINE TURBINE.

Barnaby's papers on the trials of H.M.S. *Daring,* but they are very inconclusive in many ways. Fig. 1 is submitted with much diffidence, merely as an illustration of the values of this limiting pressure.

The maximum peripheral speed of tip ever used, I think, was 12,400 feet per minute, in H.M.S. *Viper;* in H.M.S. *Velox* it is about 11,650, and in the *Londonderry* about 11,760; but many vessels have been below 9,000, which is quite normal for ordinary destroyer practice.

The pitch ratio for turbine propellers has been purposely made considerably finer than usual. Thus, the pitch ratio for the *Emerald* was about .6; in Channel steamers and cruisers of from 18 to 25 knots it has varied from .8 to 1, and in torpedo craft from 1 in H.M.S. *Velox* to 1.35 in H.M.S. *Viper,* and 1.6 in H.M.S. *Cobra;* the latter vessels having 1, 2 and 3 screws per shaft respectively, driven by identical turbines, the approximate revolutions at full speed being 900, 1,200 and 1,050, for 27, 36 and 31 knots respectively on trial.

The slip has varied from 23 percent in H. M. S. *Viper* down to about 14 percent in the *Viking*—channel steamers usually having from 17 to 24 percent. For large ocean-going vessels, about 16 to 20 percent may be used with due regard to other considerations of propeller efficiency. The section of the blades should be carefully designed in order to try to obtain a shape that will enable as high a mean pressure as possible to be adopted. Recent experiments in the model tank at Washington, D. C.,* seem to show that a symmetrical section will materially increase the pressure at which cavitation commences, and also demonstrate that in fine-pitch, high-speed screws the back of the blade should receive almost as much attention as the face. This, as the author is well aware, is no new idea, but there have been repeated indications, especially in the trials of ordinary torpedo boat destroyers, that while the gain may not be great, it is sufficient to merit attention. Mr. Parsons has advocated a 10 percent reduction of pitch at the blade tip in order to avoid excessive local thrust, which might induce early cavitation; but there seems to be no advantage in departing from a true screw. The tendency of late years, in reciprocating engine practice, has been to increase the ratio of projected disk area from the .2 of Froude's classic screw and the .22 to .26 of naval practice, to about .33; destroyer practice was included between this and .37, or even .4, at which point turbine practice may be said to commence. In this even from .5 to .56 has been used, but beyond about .58 blade interference becomes excessive, and to obtain greater area a larger diameter must be used.

The best form of blade is still undetermined. Experience seems to show that the almost circular shape, with the area disposed symmetrically on each side of the center line, and with the generating line of the screw at right angles to the axis, gives as good results as any form.

I find that the following formula will give the diameter of a turbine propeller with considerable accuracy when the effective thrust along the shaft is known, and this must be calculated in any case if the steam balance of the turbine is to be good:

$$\text{Diameter of propeller in feet} = \sqrt{\frac{\text{Effective thrust in lbs.}}{\text{Coefficient}}} = \frac{\sqrt{T}}{C}$$

This coefficient has been deduced from the limiting pressure per square inch, and the ratio of projected to disk area, and is given in diagram form in Fig. 2, where the full coefficient 400—900 is given in the left-hand scale, and values of C or $\sqrt{\text{Coefficient}}$ appear in the right-hand margin. The square root is extracted only for simplicity, whereby such coefficients as 30 are obtained for H.M.S. *Viper;* while for the *Manxman* that for the center screw is 26.4 and for the wing screws about 28.75. For large ocean-going vessels with lower designing pressures these values will be rather less, perhaps about 22.

*See D. W. Taylor's paper, Marine Engineering, February, 1905, page 72.

The following table gives a few propeller dimensions and the corresponding coefficients, which the author trusts will be of use in designing high-speed screws:

TABLE I.—PROPELLER DIMENSIONS.

Vessel.	No. of Screws.	Diameter. Ft.	Diameter. Ins.	Pitch. Ft.	Pitch. Ins.	Pitch Ratio.	Speed of Tip. Ft. per min	C (Approximate.)
Turbinia....	{9 {3	1	6	2	0	1.33	10,860	28*
		2	4	2	4	1	.....	31.3*
Viper.....	8	3	4	{4 for'd {4.6 aft.		1.2 {1.35	12,350	30.1
Amethyst....	3{1 {2	6	6	{6 {5	6 10	1 .898	9,200 {10,000	30.8
Manxman...	3{1 {2	6	2	5	7	.906	10,270	26.4
		5	7	5		.896	10,760	28.75
Londonderry	3{1 {2	5		4	6	.9	{10,550 {11,810	30.8
Dieppe.....	3	5	3	...		....	{10,100 {10,400	29
Carmania...	3	14		13		.928	8,125	21
Victorian ..	3	8	3	...		....	7,150	24.65

* These C values are calculated from the same effective thrust in each case

Compared with above values of C, reciprocating engine practice gives such figures as: R.M.S. *Lucania,* 16.5; H.M.S. *Diadem* class, 17.5; H.M.S. *Exmouth* class, 19; and standard 30-knot destroyers of 6,100 horsepower, 20.8; all of which have much lower ratios of projected to disk area, and therefore a large diameter and smaller C for a given power. Regarding these figures, it must be understood that C is an approximation only, and, owing to the difficulty of obtaining the actual power in each shaft in every case, cannot be considered as absolutely accurate. Probably, however, the error involved is under 2 percent. It seems likely, to some at present undeterminable extent but within narrow limits for each class of vessel, that the propeller efficiency is proportional to the coefficient C, and this seems to be borne out by the trials of the *Manxman* and the *Londonderry.*

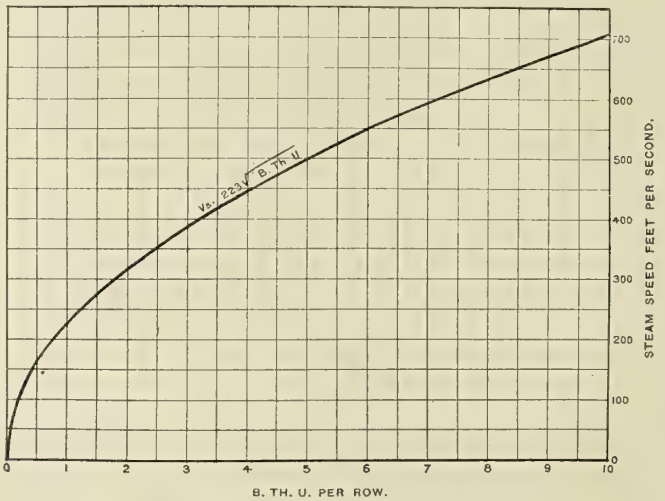


FIG. 5. DETERMINATION OF NUMBER OF ROWS OF BLADES.

Effective thrust is a somewhat subtle subject, and our knowledge of propulsive efficiency is by no means what it ought to be. These considerations will undoubtedly be brought into far greater prominence in the near future, and it is by no means improbable that the Admiralty or certain private owners will require some definite standard in this, just as coal or steam consumption is regulated at present. The more propeller efficiency is studied and understood the greater will be the improvement in the design of turbine installations for marine work; the turbine itself is a comparatively secondary consideration, and while at present propeller dimensions for turbine steamers can be quite as closely determined as those for ordinary work, the exact proportions must necessarily remain largely subject to modification from actual experience.

While a general tendency has been very noticeable toward increasing the propeller diameter and reducing the revolutions, there will of course be some point, at present undetermined, at which the triple screws used in turbine work will be distinctly less efficient than ordinary twin screws. Very largely, this is the case at present with triple screws, driven by piston engines, on account of excessive thrust deduction and interference, but probably before this point is reached the weight of the turbines will have prevented its adoption.

Having obtained the diameter of the propeller and the revolutions possible, the design of the turbine can then be undertaken, but for this no formulæ exist at present, such as are met with in reciprocating engine practice.

Restricting attention to the design of the Parsons type of turbine, a few notes on the action of the steam among the blades may be of interest. Expanding through a definite range of temperature and pressure, steam exerts the same energy, whether it issues from a suitable orifice or expands against a receding piston. Two transformations of energy take place in the steam turbine—first, from thermal to kinetic energy; secondly, from kinetic energy to useful work. The latter alone presents an analogy to the hydraulic turbine, the radical difference between the two lying in the low density of steam compared with water, and the wide variation of its volume under different temperatures and pressures.

Fig. 3 gives a sectional elevation of a marine turbine blading arrangement, and though this is only for a high-pressure cylinder the principle is exactly the same throughout. The expansion, which is approximately adiabatic, is carried out in this annular chamber from A to B, which essentially resembles a simple divergent steam nozzle, but with this difference, that whereas in a nozzle the heat energy of the working steam is expended upon itself in producing high velocities, in Parsons' turbine the total expansion is subdivided into a number of steps, in each of which a certain dynamic relationship between jet and vane is maintained. The expansion of steam at any one stage is typical of its working throughout the turbine. Each stage consists of a ring of stationary blades which give direction and velocity to the steam, and a ring of moving blades that immediately convert the energy of velocity into useful torque. The total torque on the shaft is due to the impulse of steam entering the moving blades and to reaction as it leaves them, this process being repeated throughout the turbine.

Leakage past the revolving portion of the spindle at D is almost entirely prevented by the ingenious form of frictionless packing. The fine clearances and the sudden increase of section have the effect of alternately wire-drawing and expanding the steam, so that at successive grooves it becomes increasingly difficult for the steam to leak past the clearances. In the astern turbines, a radial form of packing, depending on fine tip clearances, must be adopted, owing to the difference in expansion between spindle and cylinder. Numerous varieties of these forms of packing exist, some of them being extremely efficient in their action.

The laws governing the best theoretical velocity of steam and blades are similar to those for water turbines, but in practice some modification is necessary, and the best ratio of blade speed and steam speed is still a matter of opinion. The ideal condition for impulse turbines occurs when the peripheral velocity of the buckets is one-half that of the jet, or in reaction turbines, when it is equal to it.

Parsons' turbines, however, have been built with Vt, Fig. 4, varying from .25 to .85 of Vs, where Vt represents blade velocity at mean diameter, and Vs the steam speed due to expansion across the row in question. A very usual ratio in electrical work for large units has been $\frac{Vt}{Vs} = 0.6$; but this involves a greater number of rows than is possible in marine work, and the ratio must be reduced. These ratios need very careful calculation. The steam consumption must be accurately known in order to proportion them correctly throughout the turbine, and the necessity (which is inevitable with the present form of calking piece) of having the same area of openings in so many rows while the steam volume increases so rapidly that it adds to the difficulty of close calculation. The potential energy of the steam, corresponding to the "head" in water turbines, can easily be calculated for given pressure differences.

$$B.\,T.\,U. \times 778 = \text{Energy in foot lbs. per lb. of steam} = e.$$

$$Vs = 8\sqrt{e} = 223\sqrt{B.\,T.\,U.}$$

For a given blade velocity, it is obvious, then, that the speed ratio between jet and vane must affect the number of stages, and the greater the ratio of Vt to Vs the greater will be the required number of rows; that is, to obtain the required Vs at each stage a smaller pressure drop per row is necessary, or *vice versa*.

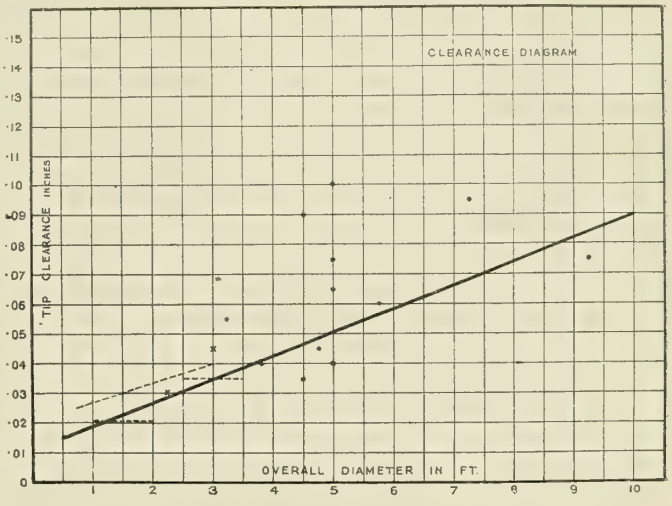


FIG. 6. DIAGRAM OF CLEARANCES AT TIP OF BLADES.

The best blading arrangement, scientifically and commercially, is the result of much theory and practice. The mean diameter is an arbitrary dimension, capable of wide variation without affecting the efficiency, provided that the number of rows is correct; it is found by assuming, from experience, a blade velocity, whence—

$$\text{Mean diameter in inches} = \frac{\text{Blade velocity in feet per sec.} \times 228}{\text{R. P. M.}}$$

To arrive at the corresponding number of rows, the revolutions being given, the ratio of Vt to Vs must be settled, from which the steam speed can be obtained; it is a convenient assumption at the beginning of any design to consider the turbine as parallel throughout and of constant efficiency, and to design on this basis. The number of rows N on one diameter can be found by working out the B. T. U. necessary to give a certain steam speed at each row (see Fig. 5). The available energy divided by the energy it is desired to abstract at each row will give the number of rows required. This result may be arrived at by various ways, but the principle involved is the same in each case. Numerous empirical coefficients for approximating steam speeds and the corresponding number of rows are obtainable from experience, and are similar in use and value to the Admiralty coefficient; that is, while they represent a crude method of doing something that should be done more scientifically, they are very simple and capable of rapid handling. Being, however, based on long and costly experiments, much reticence is observed regarding their publication. Varying, of course, with the steam pressure and vacuum, the number of rows on one diameter would involve an excessive length of turbine and also inconvenient

blade heights. It is, therefore, usual to divide the rotor into three or more stages, which have the advantage of shortening the turbine and reducing the number of rows. If n = the fraction of power developed in the first cylinder or barrel, $\frac{N}{n}$ = number of rows in the first barrel, and with the alteration of diameter and increase of blade velocity in the succeeding stages, the number of rows on other barrels is so altered as to keep, for equal powers and efficiencies:

$$(\text{Blade velocity})^2 \times \text{No. of rows} = \text{Constant.}$$

The vane speeds adopted in practice vary considerably; for some time 100 feet per second was regarded as a standard for the first row, and I think the Westinghouse Company at Pittsburg was first to make a radical departure in this and adopt far higher speeds. The maximum vane speed used for Parsons' blading is, as far as the author is aware, about 375 feet per second in the low-pressure blades, and 170 in the high-pressure blades of electrical turbines; the lowest speeds used are in marine work, and are only about one-third of these. To some extent blade speed is governed by blade height; the speed should be so modified that this may be at least three percent of the mean diameter, to reduce the proportion of clearance losses. Leakage over the tips of the blades is perhaps not so detrimental on account of actual leakage loss as in its superheating effect on steam between the row past which it leaks and the last row, because this reheating effect upsets calculations regarding openings by increasing the steam volume, and thereby affects the fluid efficiency. This leakage over the tips must be taken into account in designing reaction turbines. Temperature and diameter influence the clearance, and the stiffer the cylinder is to resist distortion due to heat the less it may be made. A clearance diagram based on measurements off a large number of machines is given in Fig. 6.

In Table II, the vane speeds adopted in various classes of work are given, and the reduction in peripheral speed on account of the propeller reducing the revolutions, and the necessary proportion of blade height modifying the diameter may be clearly seen. To this combined action is due the fact that only in the faster classes of vessels, or in those small types in which some propulsive efficiency can be sacrificed, is the turbine applicable. In slow cargo steamers, though the revolutions may be high enough, the power required is not sufficient to enable a reasonable blade height to be adopted, and it is this consideration, viz., proportion of leakage over blade tips—that curtails the wider adoption of this type of turbine. For the same low peripheral blade speed, other types of turbine are unsuitable on account of the impossibility of reducing the steam velocity sufficiently without abnormal weight and inefficiency.

Type of Vessel.	Peripheral Vane Speed.		Mean Ratio of $\frac{Vt}{Vs.}$	Number of Shafts.
	H. P.	L. P.		
High speed mail steamers....	70- 80	110-130	.45 to .5	4
Intermediate mail steamers...	80- 90	110-135	.47 to .5	3 or 4
Channel steamers.....	90-105	120-150	.37 to .47	3
Battleships and large cruisers.	85-100	115-135	.48 to .52	4
Small cruisers.....	105-120	130-160	.47 to .5	3 or 4
Torpedo craft.....	110-130	160-210	.47 to .51	3 or 4

The smallest size of marine turbine is usually larger than the average electrical turbine as far as power is concerned, and therefore does not meet with the same commercial considerations as the smaller sizes of the latter type. These are not designed for the same internal efficiency as the larger machines, chiefly on account of manufacturing cost, and they do not attain anything like the same efficiency compared with the Rankine cycle.

Speaking in reply to the discussion on his paper to the Institution of Naval Architects in 1903, Mr. Parsons said that "for all practical purposes, while the steam is traversing each set of blades as shown it behaves like an incompressible fluid, just like water would do, as the expansion is very small at each set. The frictional losses and the eddy-making losses would be practically identical within small limits with what they would be with water, and the actual forces would be in proportion to the density of the medium. * * * In the turbine blades themselves the efficiency is between 70 and 80 percent."

Using this hydraulic analogy enables us to calculate the number of stages required in a different manner; the "equivalent head," due to the steam pressure, may be found, together with that at each row necessary to give the required velocity, from which both the number of stages and the coefficient of expansion at each stage may be worked out.

In the early marine designs, such as the *Queen Alexandra* and H.M.S. *Amethyst,* the turbine drums were all made of the same diameter, and the higher speed necessary on the low pressures was got by running at considerably higher revolutions than on the high pressure shaft; but, following up the increase in propeller efficiency found to be due to the use of larger screws, the speed for each shaft is now more nearly equal, while the wing drums are made larger in diameter. The vagaries of the following wake, however, necessitate slightly different propeller dimensions on each shaft, or else slightly different revolutions with the same screws; and it is noticeable in a triple-screw arrangement, the center screw being right-handed and the wing screws revolving outward, that the starboard propeller is influenced by the center one, and almost invariably revolves at a lower speed. In a four-shaft design, due to the varying wake values at different speeds, and possibly, also, to some unequal distribution of power, the outer screws run slower at low speeds and faster at high speeds than the two inner shafts, but exact data as to this, and the possibility of allowing for it in the design, are still wanting.

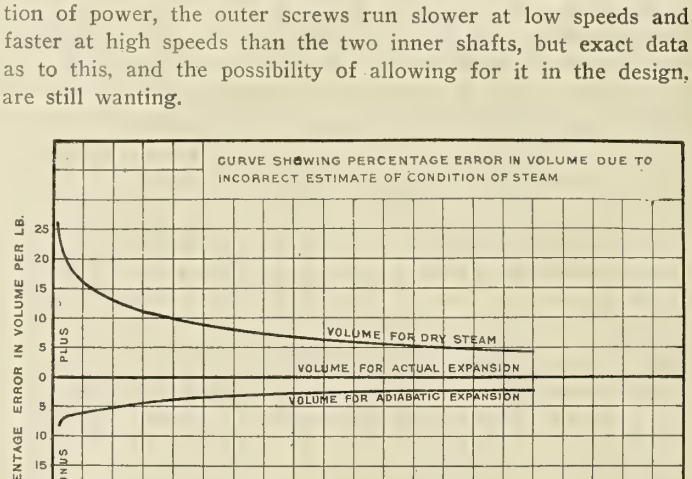


FIG. 7. CORRECTION FACTORS FOR CONDITION OF STEAM.

In all types of turbines—Parsons, Rateau, Curtis, etc.—a certain ratio must be maintained between the blade velocity and steam velocity; and as steam acquires very high velocities by expansion, the blade velocity must be maintained either by the revolutions or by large diameters, or both. As the weight increases very rapidly with the diameter, and extraordinarily so with the reduction in rotative speed, it is preferable to increase, if possible, the revolutions or the number of stages rather than the diameter, and especially should this be done in cases where, as in the Rateau or Zoelly types, the weight increases more rapidly in inverse proportion to the revolutions per minute and the diameter than it does with other types. To increase the revolutions, it may be necessary to increase the number of shafts and propellers, thus reducing the power per shaft and the effective

thrust through each screw. Increasing the diameter of the turbine adds largely to the constructional difficulties, especially of the cylinder.

Having obtained the number of rows and the diameter, the blading arrangement can be worked out in detail. The height of blade depends on the volume of steam and the speed at which it is to flow, and also on the ratio of the area of exit openings between the blades to that of the annulus between spindle and cylinder, which is about one-third in normal blades. The necessary clear area to pass the steam being equal to volume ÷ velocity, and knowing this annular factor, say 3, for a ratio of one-third (or 2 for ½, etc.), then—

$$\text{Height of blade in inches} = \frac{\text{Clear area in square inches} \times 3.}{\text{Mean circumference in inches.}}$$

The ratio of blade height to mean diameter should not be less than 3 percent or more than 15 percent, because in the former the leakage will be excessive, and in the latter the bending moment on the blade becomes too great, and the radial divergence of the blades too much. The width of blade, the shape of section adopted, and the circumferential pitch, are standard considerations, and affect the factor 3 given above. It is not proposed to enlarge upon them in this paper. It may, however, be remarked that for $\frac{Vt}{Vs}$ greater than .6 the usual shape of Parsons' section as shown in Fig. 3 should be modified to a somewhat different form of blade, with a sharper entrance edge. This section is not to be recommended, as, owing to the necessity of strengthening the blade sufficiently, the metal must be placed nearer the exit edge, thus increasing the angle between the face and the back of the exit edge of the blades, and giving, in fact, an inferior shape of opening compared with that obtainable with a blade section adapted to ratios under .6. While the axial spacing must be kept down to reduce the length of drum, it must be sufficient to allow for some play in overhauling, and sufficient clearance can be allowed here without affecting the economy. The openings between the blades to allow of the passage of the steam are very important, and must be carefully designed. The actual volume of the steam—not the volume per pound, as found in tables, or the volume due to adiabatic expansion, but the exact volume per pound at any point along the turbine—must be determined, in order to arrive at the desired adjustment of velocities. It is extremely doubtful whether the present blading arrangements give the best results; greater accuracy of calculation and consequently improved pressure distribution and efficiency seem likely to follow the use of a more mechanical blading construction.

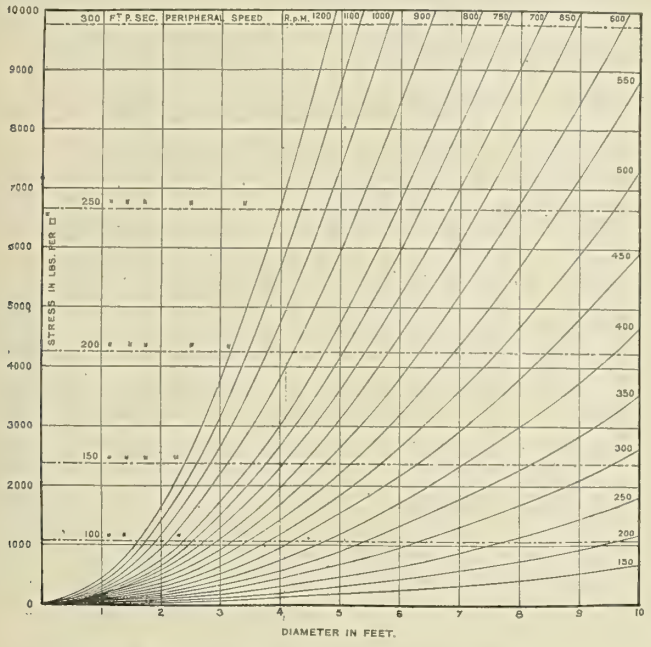


FIG. 8. STRESSES DUE TO CENTRIFUGAL FORCE.

Fig. 7 shows the percentage error involved in using either the dry volume or that due to adiabatic expansion, compared with the correct volume corresponding to the actual expansion in a large turbine using dry saturated steam at the first row of blades and 27 inches vacuum. Attention must be paid to the effect of approximately adiabatic expansion and the consequent moisture in the steam.

For manufacturing convenience, as well as to allow for the expansion of the steam, the blade heights are stepped up, but no rule exists for this; the blades might be of one or of sixty-four heights, provided the blade openings are correct. It is, however, convenient to step them, as in Fig. 3, say, in 8 steps of 8 each, or 4 of 16—even 9 of 7 would do—and so avoid any great variation from the annular area factor 3 as shown above. To obtain heights and areas, it is best to plot off, graphically, volumes, steam speeds, and clear areas required. The use of standard blade heights will then enable the number of stages and rows per stage to be determined; wide differences can be made in

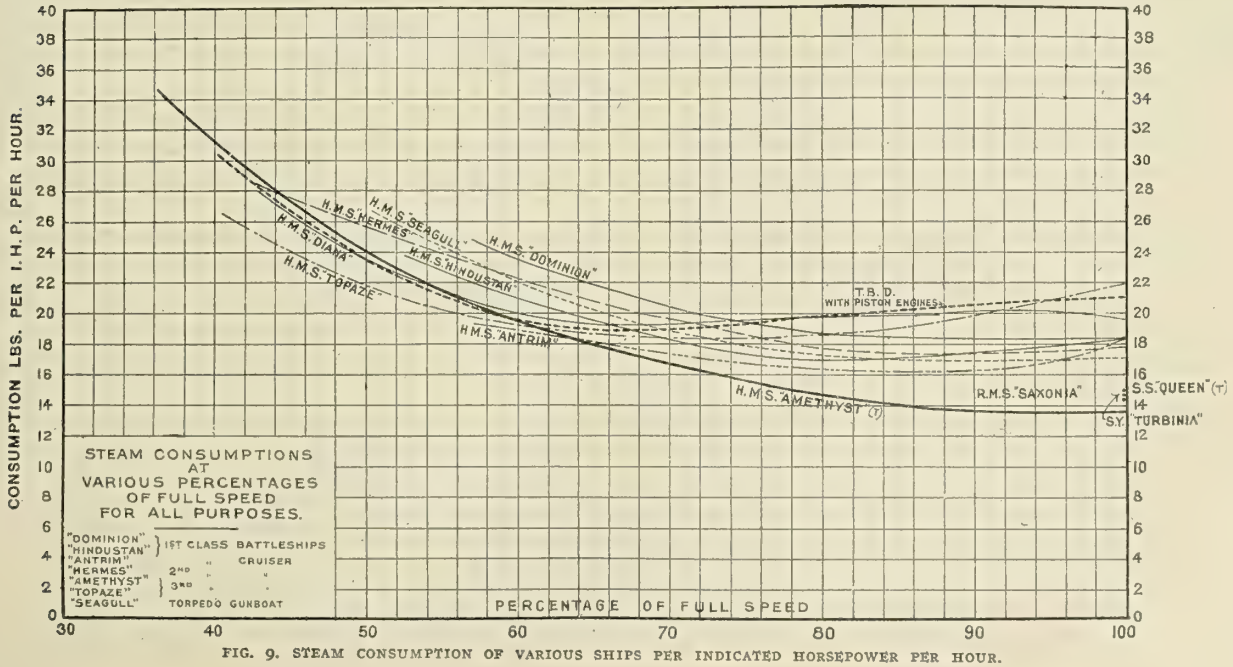


FIG. 9. STEAM CONSUMPTION OF VARIOUS SHIPS PER INDICATED HORSEPOWER PER HOUR.

any arrangement without materially affecting the economy. The best arrangement is largely a matter of convenience and experience.

The material of which blades are usually made is a mixture of cheap brass containing about 16 percent of copper and 3 percent of tin. Alloys containing zinc are extremely unreliable for high temperatures, but blades containing about 98 percent of copper have been found very satisfactory for use with high superheats. More recently a material containing about 80 percent of copper and 20 percent of nickel has been adopted, and this is undoubtedly the best blading material existing. Steel blading, drawn in the same way as the usual brass section, has been used in the United States with fairly good results. The process of drawing turbine blades gives an extremely tough skin to the metal used, not only increasing the tensile strength, but greatly decreasing the chances of erosion.

It seems probable that the usual calking piece now adopted will be discarded in favor of a machine-divided strip into which the blades may be fitted, and instead of the slotting, wiring, lacing, and soldering process at the tip, a similarly machine-divided shroud will be used, giving a far stronger construction, and enabling finer clearances and better workmanship to be obtained, at the same time considerably reducing the cost of manufacture and the risk of blade stripping.

The chief causes of the latter may be set down to bad workmanship in fixing the blades, defective blade material, excessive cylinder distortion (this is probably the most fruitful cause, and is a serious one, being due to bad design), whipping of turbine spindles (which is also due to bad design, or bad balancing), wear of bearings (which is very remote), and the introduction of extraneous substances such as water or grit. In fact, blade stripping may be said to generally occur from preventible causes.

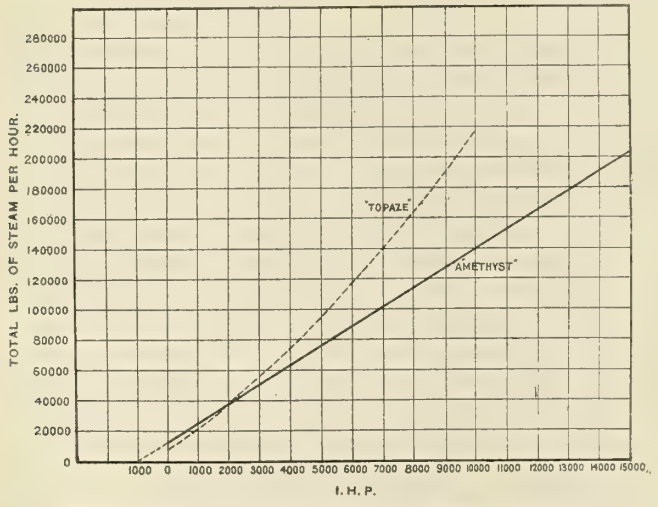


FIG. 10. TOTAL STEAM CONSUMPTION AT VARIOUS POWERS.

Small vibrations of very high frequency occasionally set up an action in certain rows of responsive length that fatigues the blade material and causes the loss of blades without any fouling at all.*

Due to the action of the steam, an end thrust occurs in the direction of the propeller, which is advantageously used in partially balancing the propeller thrust, thereby reducing the size of thrust block necessary. A margin must be allowed here, and the propeller thrust is not entirely balanced by the pressure on the annulus between the dummy-ring diameter *D*, and the spindle, Fig. 3, *C*, plus the end pressure on the blades. For the diameter *D* to give the required annulus the effective thrust, as well as that of the propeller, must be carefully calculated; and experience shows that there is a drop in steam pressure varying from 10 to 15 pounds per square inch between the pipe inlet to the high-pressure receiver and the first row of blades, which should be considered in designing this balancing area. The number of rows of dummy packing used varies according to the designer's judgment very largely, and may be modified according to the pressure and the clearance allowed—say 7-1000 to 15-1000 of an inch in electrical work and rather more in marine work.

The dimensions of the astern turbine are arrived at in the same manner as those of the ahead, the efficiency being largely sacrificed on account of weight and space; generally, the mean diameter is made practically the same as that of the high-pressure drum.

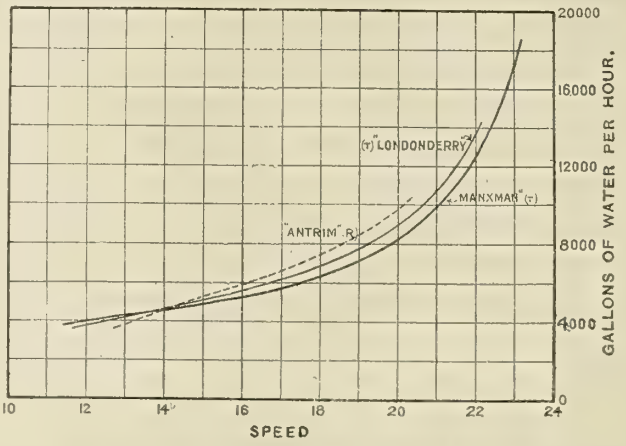


FIG. 11. STEAM CONSUMPTION OF MIDLAND STEAMERS.

To a large extent, the inferior maneuvering capabilities of the earlier turbine steamers were due to insufficient astern power.

It may be remembered that in a marine turbine the spindle is in compression and the cylinder in tension when working. In electrical turbines where the end thrust must be eliminated by the use of balancing pistons, the spindle is in tension and the cylinder is balanced. The shafts between the turbine bearings and the drum must be made amply stiff enough, as well as strong enough, for any sag in the spindle will destroy the clearance. As will be seen from Fig. 8, the stresses due to centrifugal force are very low in the Parsons turbine, and except in occasional low-pressure barrels do not exceed about 7,500 pounds per square inch, while at the high-pressure end they are usually under 2,000.

The pressure on the bearings in a turbine is due only to the weight of the spindle, plus the negligible addition in marine work of that due to any gyroscopic action; it may be taken as from 80 to 90 pounds per square inch as long as the rubbing velocity does not exceed 30 feet per second. If it does, the pressure must be reduced so that the product of pressure × velocity does not exceed 2,500 or 2,700. In land work, 50 pounds × 50 feet is very common. The friction heat of the bearings added to that due to conduction through the pedestals necessitates the use of large oil coolers, and in the case of very high temperatures, of special kinds of oil. If possible, the bearing temperature should not exceed from 140 degrees to 150 degrees F., though the writer has known of 190 degrees F. being used without trouble. In marine turbines this temperature is usually much lower. Rigid bearings are used for marine spindles, not the flexible type adopted in land work.

Space does not permit of more than passing reference to cylinders; but it would be difficult to exaggerate the importance of very careful design in this connection. Cylinders with heavy flanges on the center line distort in a very curious fashion when heated with their axis horizontal, and measurements taken off a hot cylinder on a surface plate with micrometer gauges reveal some very remarkable facts. When working, the temperature along the cylinder falls possibly from 400 degrees to 100 degrees F. in a distance of 6 or 8 feet, and, unlike the reciprocating

*The writer had experience of this early in 1905 when a 5,000-K.W. turbine, under test in one of the large New York power stations, shed several rows of blades. This difficulty has also occurred in Europe, and can be circumvented by an alteration in the position of the lacing strip.

engine, this remains constant; the radial expansion is consequently more at one end than at the other; while at any point along the turbine the tendency is to expand less at the flanges than at the top and bottom. For this reason ample clearance must be allowed; exactly what this will be when spindle and cylinder are hot is hard to say, but it seems most likely that the total clearance area will differ but little from what it is when cold.

The longitudinal expansion when hot is often very marked, and in all turbines necessitates provision for the resultant movement at one end. In marine work the after end of the cylinder is secured to the vessel, the engine seating also performing the function of a thrust-block seat, while the forward end slides forward, taking with it the entire shafting. The thrust block is at the

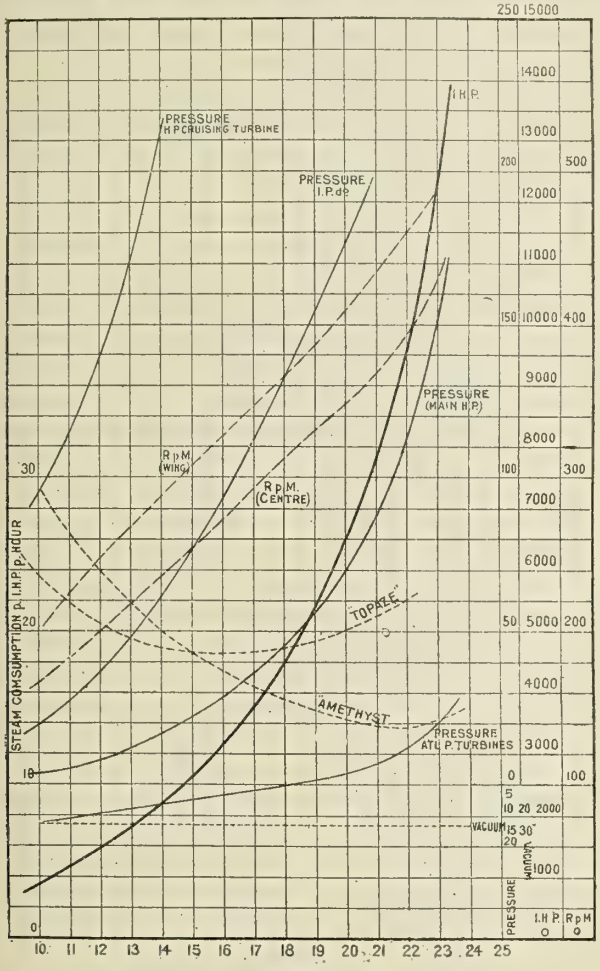


FIG. 12. RESULTS OF TRIALS OF AMETHYST.

forward end of the cylinder, and also performs the duties of an adjustment block for setting the longitudinal clearances, to do which generally necessitates uncoupling the shafting abaft the turbine.

The difference in expansion between the cylinder and spindle, from the thrust block to the dummy ring, may be the cause of serious difficulties in large marine turbines, unless the closest attention is paid to this feature in the design; and "warming up" with these large cylinders needs possibly even more care than is essential with large piston engines.

On shipboard the turbine cylinders are practically under one's feet, and the radiation from them is very unpleasant, especially if there is any leakage from the glands. To all who are responsible for the lagging of cylinders and the system of ventilation in turbine engine rooms I would call attention to the possibility of their having to stand a watch of 4 or 6 hours on the top of the high-pressure cylinder, such as is the case in the *Eden* or *Amethyst,* the heat in the latter vessel being almost unbearable. With reciprocating engines one stands on a comparatively cool lower platform with the cylinders overhead, and with some chance of the hot gases rising clear, but in naval turbine work under a low deck this point has not met with adequate attention.

In the course of operation, more especially in marine work where no superheaters are used, there is a distinct tendency for the turbine to be supplied with wet steam, the effect of which on the economy is very marked. Experiments that have been made show that the percentage increase in consumption is about twice that of the moisture in the steam. For instance, with 2 percent of moisture in the steam at the first row, the consumption is increased about 4 percent.

A considerable amount of data on the performance of turbines compared with reciprocating engines for marine work is now available. The Admiralty has had tested both cruisers and torpedo-boat destroyers, exactly similar but for their engines and propellers, and trials of the Midland Railway Company's steamers and other cross-Channel boats have corroborated the results regarding economy obtained from the naval vessels. In Fig. 9 is given the steam consumption per unit of power of H.M.S. *Amethyst** compared with that of several recent warships, and it is noticeable that only below from 55 to 60 percent of their full speed does the consumption of the turbine exceed that of the piston engines. Very seldom do vessels steam below these speeds. Cruisers carrying relief crews to the China or Australian stations usually proceed at about 60 percent of full speed, and in the Atlantic maneuvers of 1903 nearly 80 percent of full speed was maintained by the large fleets, while the Japanese battleships built in England made their first voyage to Japan at about 54 percent of their full speed; at which ratio the consumption per indicated horsepower of both the *Hindustan* and *Dominion,* representing very recent battleship construction by eminent builders, is materially in excess of that of the first installation of warship turbines (not including the destroyers). The total consumption of H.M.S. *Amethyst* and *Topaze,* plotted on a base of power, is given in Fig. 10, while Fig. 11 shows same for the Midland Railway boats.† The progressive trials of H. M. S. *Amethyst* are shown in Fig. 12, and, in view of the results obtained from these various vessels, the wholesale adoption of turbine machinery in the royal navy is not surprising.

It is probable that the adoption of cruising turbines will be discontinued before long, and this view seems to be corroborated by the consumption trials of the Midland Railway steamers. Down to 60 percent of her full speed the *Manxman* required less water than the highly efficient *Antrim;* and with a different blading arrangement in the main turbines such a result could be equaled, if not improved on, in war vessels. The additional complication involved with two cruising turbines and their accompanying leakage and receiver losses, together with a considerable increase in weight and space occupied, largely modifies any advantages obtainable in the way of reduced consumption at lower powers. An improved (and easily obtainable) design of main turbine blading should give a better result at the highest powers, practically the same at intermediate powers (as in the case of H.M.S. *Amethyst,* from 14 to 20 knots), and only slightly inferior at speeds below 14 knots; while it will undoubtedly admit of greater ease of handling and be much simpler. In a triple-shaft arrangement the unequal distribution of power on the wing shafts, due to the use of cruising turbines, is a distinct disadvantage: the fluctuation in rotative speed, due to shutting off the high-pressure cruising turbine, may be seen from the trials of H.M.S. *Amethyst.*

*Since the above-mentioned results were obtained, the steam piping has been altered so as to permit the auxiliary exhaust steam to pass through the main low-pressure turbines when desired. This arrangement considerably decreases the consumption at low speeds, bringing the *Amethyst's* consumption below that of her sister ships down to ten knots, or about 45 percent of full speed.

† Fig. 11 is compiled from the results given in Mr. Gray's paper to the Institution of Naval Architects, July, 1905.

In conclusion, the writer would remark that it is impossible in the scope of a paper such as this to touch more than lightly on a subject which is of such vast importance. Many of the points dealt with above, such as cavitation, blading, cylinder design, etc., would require a volume to describe. While turbines, perhaps, are still in their infancy, they are already largely supplanting the reciprocating engine in many types of vessel. The next few years will undoubtedly show as great an improvement as has taken place since the advent of the *King Edward,* barely five years ago, the more especially as the subject will, henceforward, be engaging the attention of all engineers instead of a few specialists; and that this improvement will more than justify the policy of the Admiralty and of the Cunard Company is already certain.

EXPERIMENTAL RESEARCHES ON THE PERFORMANCE OF SCREW PROPELLERS. *

BY W. F. DURAND.

In the fall of 1897 a preliminary paper on this subject was read before the Society, giving the results of certain tests made on a series of seven model propellers, all of the same pitch ratio and form of blade, but with varying widths, thus giving a series with area ratio varying between the limits of .18 and .72. Further prosecution of the work was delayed by various causes, but at length it became possible under much improved conditions in the canal of the hydraulic laboratory of Cornell University, and during the past two years, through the aid of a grant from the Carnegie Institution of Washington, D. C., the work has been carried forward in the intervals of other duties of the observers and other uses of the canal. At the present time the original program (including the examination of several series similar to that first reported on and with selected pitch ratios covering the field of normal design) has been completed, together with certain special and auxiliary investigations not coming within the scope of the present report. The propellers used in the first investigation have also been included in the general program carried out with the improved means made possible by the experimental canal, so that the present report covers in effect the performance of 49 model propellers distributed over the field of pitch and area ratios, as shown by the following table:

PITCH RATIOS.

Area Number.		.9	1.1	1.3	1.5	1.7	1.9	2.1
2	Area Ratio.	.18	.18	.18	.18	.18	.18	.18
3		.27	.27	.27	.27	.27	.27	.27
4		.36	.36	.36	.36	.36	.36	.36
5		.45	.45	.45	.45	.45	.45	.45
6		.54	.54	.54	.54	.54	.54	.54
7		.63	.63	.63	.63	.63	.63	.63
8		.72	.72	.72	.72	.72	.72	.72

The diameter of all propellers was 12 inches and the diameter of the hub 2.4 inches. The shape of all blades was elliptical, the maximum width in tenths of the radius being expressed by the area number in the table.

The experiments were carried out in the canal of the hydraulic laboratory at Cornell University. This canal or tank is about 340 feet in length by 16 feet in width and 10 feet in depth. The canal is provided with water from the reservoir above, through double shut-off gates and an intermediate lock or measuring chamber. Once filled the water is quiescent, and the level may be maintained where desired by an adjustable weir at the western end.

*Read at the thirteenth general meeting of the Society of Naval Architects and Marine Engineers, November 16, 1905.

The principal items of the equipment of this canal are as follows:

(1) A carriage or truck spanning the canal and carrying the propellers to be tested and the apparatus required, and running on a track extending the length of the canal.

(2) A transmission dynamometer for the measurement of the power absorbed by the propeller.

(3) Means for driving the propeller through the dynamometer at speeds varying over a wide range, and for any run in constant proportion to the speed of the truck along the rails.

(4) A thrust dynamometer for measuring the thrust developed.

(5) Means for registering time, distance and revolutions.

GENERAL DISCUSSION.

For the discussion of the results we shall use the following notation:

T = thrust.
Q = torque or turning moment = work $\div 2\pi N$.
d = diameter of propeller.
p = pitch of propeller.
N = revolutions per minute.
s = slip ratio.
u = speed in feet per minute = $pN(1-s)$.
e = efficiency.
a = factor in expression for thrust which depends on slip ratio.
b = factor in expression for thrust which depends on pitch ratio.
c = factor in expression for thrust which depends on area ratio.
x = factor in expression for torque which depends on slip ratio.
y = factor in expression for torque which depends on pitch ratio.
z = factor in expression for torque which depends on area ratio.

Then quite independent of the details of propeller theory we may put

$$T = d^2\,(pN)^2 \times abc = d^2p^2N^2abc, \quad \ldots\ldots\ldots\ldots (1)$$

$$Q = d^2 \quad (pN)^2 \times xyz \div N = d^2p^3N^2xyz. \quad \ldots\ldots\ldots (2)$$

Equation (1) is simply equivalent to the statement that the thrust developed by a propeller will vary with the square of the

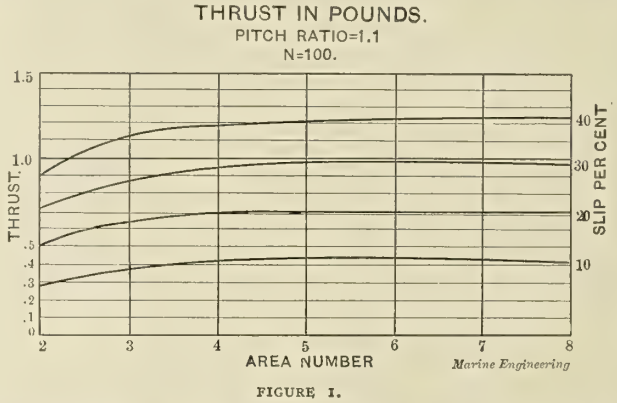


FIGURE 1.

diameter and with the square of the speed, and that it will depend on the slip, pitch ratio and area ratio. The performance will furthermore depend in chief measure on the shape of the blade and on the amount and distribution of the thickness. Since, however, the present experiments provide no means of analyzing out any such influence factors, they must be considered as constant, corresponding to the uniform elliptical shape of the blades and to the standard schedule of thickness employed. In other words, the three items in these experiments which were subjected to determinate control, and which must, therefore, serve as the variables in terms of which the results may be expressed, are slip, pitch ratio, and area ratio. The formulation of (1) expresses, therefore, in algebraic language the universally accepted relation of thrust to diameter and speed, together with the general fact of dependence on these three variable items, through the factors represented by a, b, and c.

In a similar manner (2) is equivalent to the statement that

the work absorbed by the propeller will vary as the square of the diameter and as the cube of the speed, and that it will be dependent also on slip, pitch ratio, and area ratio; and, furthermore, that torque varies as the work divided by the revolutions.

We then have:

$$T \times pN\,(1-s) = \text{useful work} \quad \ldots\ldots (3)$$

$$Q \times 2\pi N = \text{total work} \quad \ldots\ldots (4)$$

$$e = \frac{TpN\,(1-s)}{2\pi QN} = \frac{Tp\,(1-s)}{2\pi Q}$$

or

$$e = \frac{(1-s)\,abc}{2\pi xyz}.$$

Thus far the equations deduced are general. Next, to apply them to the present case, we shall take 100 r. p. m. as a standard

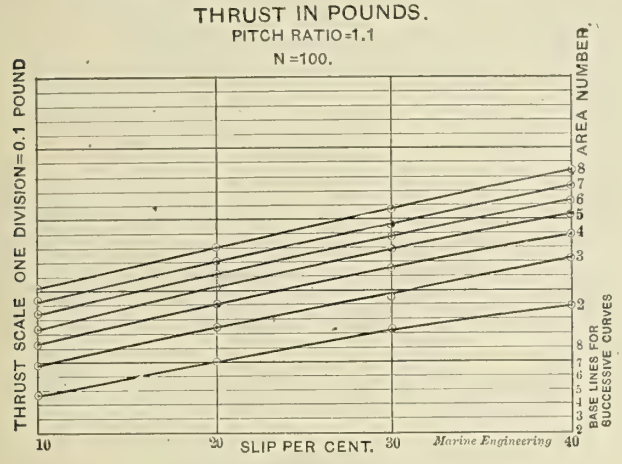


FIGURE 2.

or unit number of revolutions. Then, remembering that d in all cases equals one foot, we shall have in (1) and (2) for the standard revolutions:

$$T = p^2abc. \quad \ldots\ldots (7)$$

$$Q = p^3xyz. \quad \ldots\ldots (8)$$

Now equations (1) and (2) show that for fixed values of d, p, a, b, c, x, y, and z (that is for a given propeller at fixed value of the slip) T and Q will vary as N^2, and hence the actual observations which were made at revolutions varying from 200 to 600 may be reduced to the standard revolutions by dividing by the square of the r. p. m. expressed in units of 100. This gives the analytical foundation for the step in the process of reduction by which the actual values of T and Q are reduced to the standard revolutions of 100 per minute, and thus made ready for plotting.

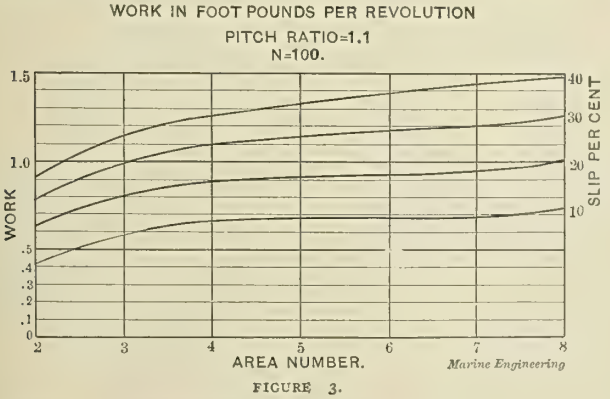


FIGURE 3.

Inversely it follows that if the values of a, b, c, x, y, z, or of their products abc and xyz as derived from a given observation, be assumed to hold for a larger propeller of similar form at the same slip, then (1) and (2) will serve to compute the thrust and torque, and hence the useful work and total work which the operation of such a propeller at revolutions N would involve. This gives the basis of the application of such results to the operation of design, a topic which will not be considered in detail in the present paper.

As a different basis for the reduction of the observed values to standard conditions we may write (1) as follows:

$$T = d^2\,[p^2N^2(1-s)^2]\,\frac{a}{(1-s)^2}\,bc.$$

But

$$pN(1-s) = u,$$

and we have, therefore:

$$T = d^2u^2\,\frac{abc}{(1-s)^2} \quad \ldots\ldots (9)$$

Similarly we find:

$$Q = d^2pu^2\,\frac{xyz}{(1-s)^2} \quad \ldots\ldots (10)$$

For a standard unit speed of 100 f. m. = 1 unit and with $d = 1$ we then have:

$$T = \frac{abc}{(1-s)^2} \quad \ldots\ldots (11)$$

$$Q = \frac{pxyz}{(1-s)^2} \quad \ldots\ldots (12)$$

These equations show that for fixed values of d, p, a, b, c, x, y, z (that is, for a given propeller at fixed value of the slip), T and Q will vary as u^2, and hence the actual observations which

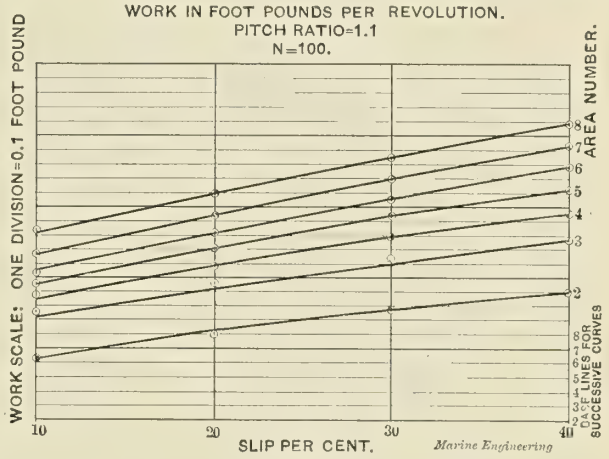


FIGURE 4.

were made at a nearly constant speed of 300 f. m. may be reduced to the standard speed by dividing by the square of the actual speed expressed in units of 100 f. m. This gives the analytical foundation for the step in the process of reduction by which the actual values of T and Q may be reduced to the standard speed of 100 f. m., and thus made ready for plotting.

Inversely it follows that if the values of $a \div (1-s)^2$, b, c and $x \div (1-s)^2$, y, z, or of their products $abc \div (1-s)^2$, $xyz \div (1-s)^2$ as derived from a given observation be assumed to hold for a larger propeller of similar form at the same slip, then (9) and (10) will serve to compute the thrust and torque, and hence the useful work and total work which the operation of such a propeller at speed u would involve.

These two methods of reduction are readily connected together by noting that the relation between (7) and (11) or (8) and (12) is expressed by the factor $p^2\,(1-s)^2$. That is, the values given by reduction to a standard speed of 100 f. m. as in (11) and (12) when multiplied by $p^2\,(1-s)^2$ will give the values reduced to standard r. p. m. of 100 as indicated by (7) and (8).

In the reduction of the observations to suitable form for

graphical representation either method might be followed; that is, the results might be reduced to a standard value of the r. p. m. or of the speed in feet per minute. Both methods are equally correct and either will express all the facts. In the actual case it is found that the values of T and Q when reduced to a standard value of the r. p. m. and plotted for a given propeller on an abscissa of slip fall nearly on straight lines, and that in groups the resulting curves are more compact and somewhat more manageable than those resulting from reduction to standard speed. For this reason the reduction to standard r. p. m. has been preferred for the general method of treatment, and especially for the graphical representations shown in the various plates,* 5-32 expressing T and Q on abscissæ of slip and area.

REDUCTION OF DATA.

For the actual reduction of the various observations the following general routine was followed:

The initial data consisted of dynamometer records giving measures of thrust and turning moment, together with strips of paper containing the records of time, distances and revolutions. The latter were readily converted into revolutions per minute and speed in feet per minute, from which, with the known pitch of the propeller, the slip could readily be found.

The actual data were all worked over by reduction to both standard speed and standard r. p. m.; but for reasons as given above, the reduction to standard r. p. m. is preferred for presentation of the results in graphical form.

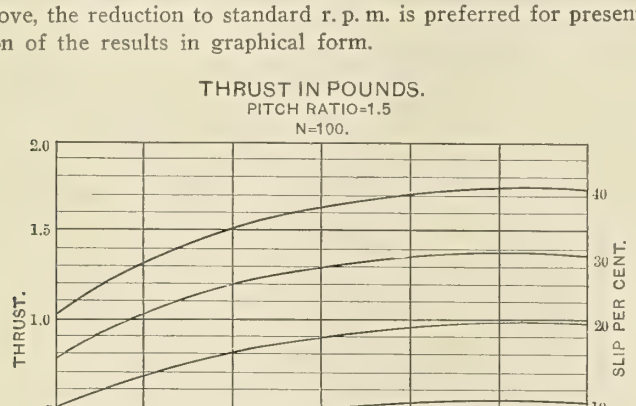


FIGURE 5.

The dynamometer records, through the usual calibrations, were readily transformed into actual forces in pounds or turning moments in foot pounds and such actual forces and moments were then reduced to their corresponding values at 100 r. p. m. by dividing by the square of the revolutions expressed in units of 100.

These results give for each propeller for both T and Q a series of values corresponding to successive values of the slip, as it may have resulted from the several settings employed, and all reduced to the standard of 100 r. p. m.

The number of such points or single observations varied somewhat with special circumstances, but was usually not less than 25 or 30 and sometimes rose to 50 or more. Such points for T and Q† were then plotted on the axis of slip and a fair curve was drawn through and among the points giving the best general graphic law between T and s or Q and s which could be derived from the observations. The resulting curves are indicated as nearly as may be, and, of course, at reduced scale, by the spots on the diagrams, such spots representing these curves at exact values of the slip 10, 20, 30 and 40 percent. Cross curves were then drawn at these values of the slip, giving the values of T and Q on an axis of increasing area. Such cross curves naturally

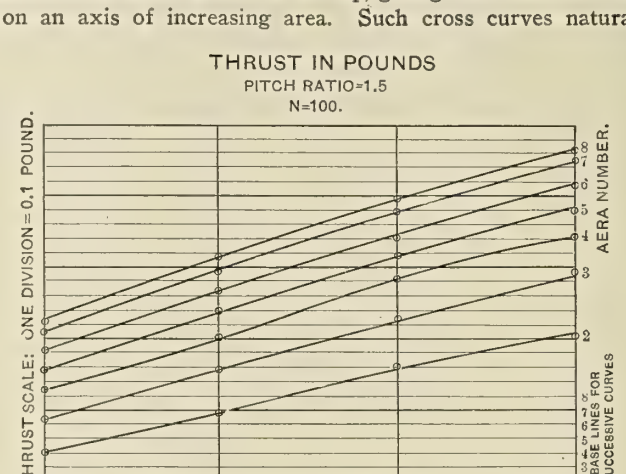


FIGURE 6.

indicated the need of some fairing up process with reference to the two sets of curves in order to derive a smooth general law for the series of seven propellers of fixed pitch and varying area. The results of such fairing up process are shown by the actual curves, one-half showing values of T and Q on slip as an abscissa and the remainder the same items with area as an abscissa.

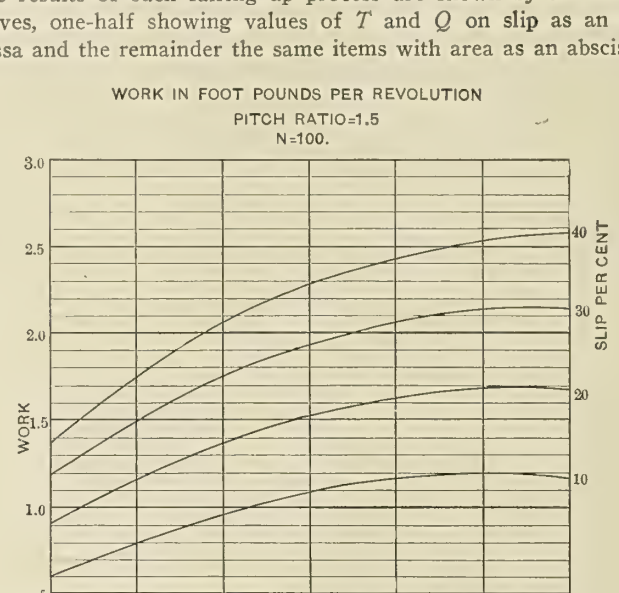


FIGURE 7.

It should be further noted that naturally the propellers as actually made were not all exactly the same diameter nor were all of exactly the rated pitch or area. All geometrical functions of the propellers as actually used were, however, carefully determined at the close of the experiments and the results tabulated. These measurements included determinations of diameter, of the area of each blade, of the pitch of each blade and average pitch, and a general map of the thickness lay out on each blade throughout the series. The variations from normal rating were in all cases quite small and for the most part might have been neglected. Appropriate corrections were, however, applied in such manner as to allow for these minor divergencies from the rated characteristics. We shall, however, consider the details of these corrections as lying beyond the scope of the present paper.

* Due to lack of space, twelve plates only have been reproduced.

† As actually plotted the values of Q are multiplied by 2π thus giving work in foot pounds per revolution.

The resulting groups of curves may, therefore, be viewed as giving the best general law connecting T and Q with slip and area, each for a given value of the pitch ratio as indicated.

EFFICIENCY TABLE.

Pitch Ratio.	Slip Ratio.	Area Numbers. 2	3	4	5	6	7	8
.9	.10	.59	.65	.65	.62	.62	.61	.60
	.20	.66	.68	.68	.69	.66	.65	.64
	.30	.66	.68	.66	.66	.64	.61	.60
	.40	.62	.61	.62	.61	.60	.59	.56
1.1	.10	.63	.63	.63	.65	.65	.63	.56
	.20	.70	.70	69	.70	.70	.66	.62
	.30	.70	.68	.68	.67	.65	.62	.59
	.40	.66	.64	.63	.62	.59	.57	.56
1.3	.10	.51	.54	.55	.59	.61	.64	.65
	.20	.66	.68	.69	.68	.68	.69	.69
	.30	.68	.68	.68	.66	.66	.66	.66
	.40	.66	.65	.62	.61	.60	.60	.59
1.5	.10	.47	.56	.61	.60	.61	.61	.62
	.20	.65	.71	.69	.69	.69	.71	.71
	.30	.71	.72	.72	.69	.69	.68	.68
	.40	.68	.69	.66	.64	.63	.62	.60
1.7	.10	.50	.58	.61	.63	.63	.64	.65
	.20	.73	.73	.72	.71	.70	.70	.70
	.30	.72	.73	.72	.70	.70	.68	.68
	.40	.70	.69	.66	.64	.64	.63	.63
1.9	.10	.46	.56	.62	.63	.61	.62	.59
	.20	.67	.71	.74	.73	.71	.71	.69
	.30	.70	.73	.73	.73	.70	.69	69
	.40	.65	.67	.67	.66	.64	.63	.63
2.1	.10	.52	.58	.61	.59	.61	.61	.61
	.20	.70	.69	.68	.69	.68	.68	.70
	.30	.74	.73	.70	.69	.69	69	.69
	.40	.68	.64	.64	.63	.62	.62	.64

Values of the efficiency were then found by multiplying the values of T by $100p\ (1-s)$ and those of work per revolution by 100, and dividing the former by the latter. The results for efficiency are shown in tabular form in Table I, such mode of presentation being preferred for the present paper on account of the large number of separate diagrams needed to show the various efficiency curves which would result.

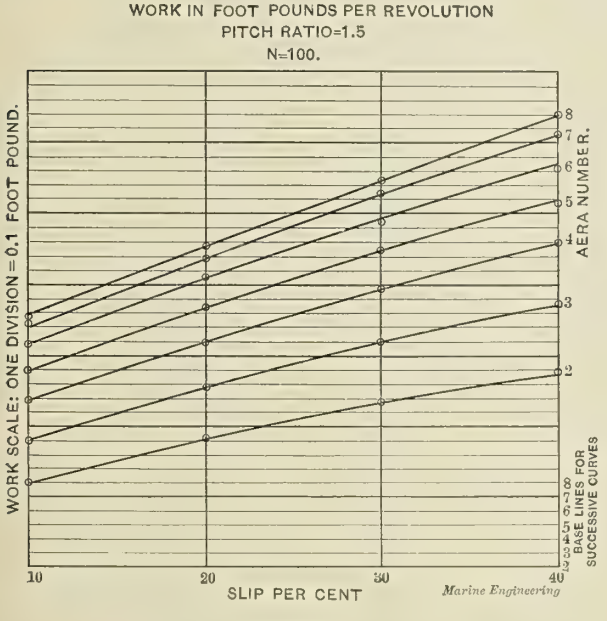


FIGURE 8.

The actual work of reduction has naturally contained a great number of special operations and devices in order to develop a comprehensive relation between the initial data and the final results in terms of thrust, work and efficiency. Into the minor details of such operations it will be unnecessary to go in the present connection, and the above brief statement will serve to show the general lines which were followed in the reduction of the results as herein given.

GENERAL RESULTS.

The curves showing values of T and Q on s, the values being reduced to standard r. p. m., are lines nearly straight or curved slightly concave to the axis of slip. This general character of the relation between Q and s or T and s is distinctively characteristic of the propeller, and has been indicated by all previous experiments carried out in detail and covering the determination of the data in such form as to bring out this relation. There is, of course, no reason for assuming that such law is naturally linear or that there is any natural reason for expecting a straight line relation between these quantities. The general trend of the curves as shown indicates clearly, however, a somewhat slower increase of T and Q with s than the exact linear law would require. It may be noted here also that further special experiments which are not included in the data given in the present paper and which show the results for 100 percent slip, indicate such values of T and Q at this extreme point as would require a marked curving over of the line, especially for values of s beyond 30 and 40 percent. From all the data at hand it can be fairly assumed that the typical relation between T and s or Q and s when T and Q are reduced to standard r. p. m. is represented by a line generally curving over concave toward the axis of s. The curvature is, furthermore, so gentle between ordinary limits of perhaps 10 and 30 percent that without serious error the law between these points may be taken as linear and, therefore, as expressible by a simple equation of the first degree.

It will be further noted that the slope of such lines increases with the area ratio and with the pitch ratio according to some complex law which is obviously a function of these two characteristics.

The relation of T to increasing area is shown. These curves show comprehensively the general relation between these two

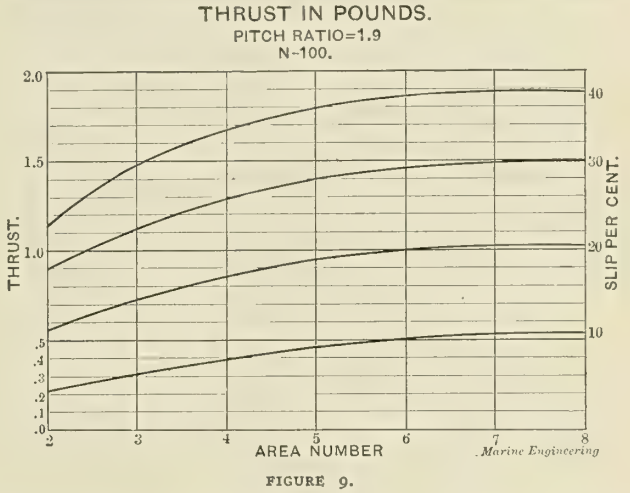


FIGURE 9.

features and indicate in particular how inconsiderable is the gain in thrust beyond area ratios of .50 or .60. Indeed, for low pitch ratios the thrust falls off slightly beyond such values of the area, showing how useless it is, especially with low pitch ratios, to increase the area beyond the usual upper limits of .50 to .60. This same general result has been indicated by other experiments, and among others by those reported by Naval Constructor Taylor at the last meeting of the Society. For the higher pitch ratios there is, as a rule, a gradual increase in thrust up to the largest employed, but the actual gain beyond area ratios of .50 or .60 is so small as to be of comparatively small importance. The general character of these curves varies but slightly with the slip, all showing the same comparatively rapid increase between area numbers 2 and 4 or 5, gradually decreasing to no increase of importance between 6 and 8.

Similarly, values of Q are shown, plotted on area ratio. These curves have all the same general character as those for T. The increase with area is, however, continuous and relatively more pronounced than for T.

These general results for T and Q plotted on area are in accord with the indications of theory on the same general points. It is clear that with a low pitch ratio in particular the effective thrust will be limited by the interference of one blade with another as the area is much increased, and by the continuous increase of the frictional or tangential component of the forces which act along the face of the blade and exert a continuously increasing longitudinal component aft, while the direct forward thrust due to the interaction of the blade and the water reaches an early limit as the blade is widened beyond limits corresponding to .40 or .50 area ratio. On the other hand the tangential component at least will grow continuously with the increase in area and this will give a small though continuous increase in Q with increase of area. With higher pitch ratios the same general relations hold, but the blades interfere less with increasing area and the limit of forward thrust due to the interaction of propeller and water is less quickly reached, while at the same time the value of Q increases continuously with area, though slowly beyond a limit of .50 area ratio.

The efficiency relations shown in the table are of high interest and the general laws indicated will repay careful attention. The general character of the variation of efficiency with slip for a given propeller is well known and is indicated by the figures of the table. The values are low for low slip, rising rapidly to the maximum in the neighborhood of 20 to 30 percent and then falling more gradually as the slip increases beyond such value.

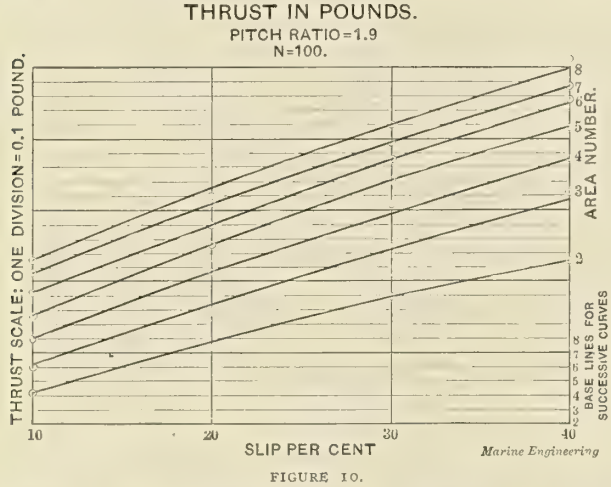


FIGURE 10.

In general there is indication that the propeller of very small area reaches its best efficiency at a relatively high slip, usually in the neighborhood of 30 percent, while that of large area reaches its best value at about 20 percent or below. The highest values seem, furthermore, to be only slightly dependent on either area or pitch ratio. The low pitch ratio with high area, as would be expected, shows the lowest values of the efficiency. Outside this territory all the propellers reach at some value of the slip a value of the efficiency closely approximate to 70, the best general results being indicated for fairly high pitch ratios and moderate areas.

Stated otherwise, these general efficiency relations indicate for a low slip, such as 10 percent, a general increase of efficiency with area, except for the lowest pitch ratios, when such increase is followed by a decrease for the highest areas. For higher values of the pitch ratio and low slip, however, there is as a rule, first a marked and then a slow increase of efficiency with increase in area, the best results being given for area ratios of 35 to 45. On the other hand, for high values of the slip, such as 40 percent, there is, as a rule, a continuous though perhaps slow decrease of efficiency with increase of area from number 2 to 8, or throughout the range. Intermediate between these extremes we find for moderate or average values of the slip between 20 and 30 percent but slight change of the efficiency with

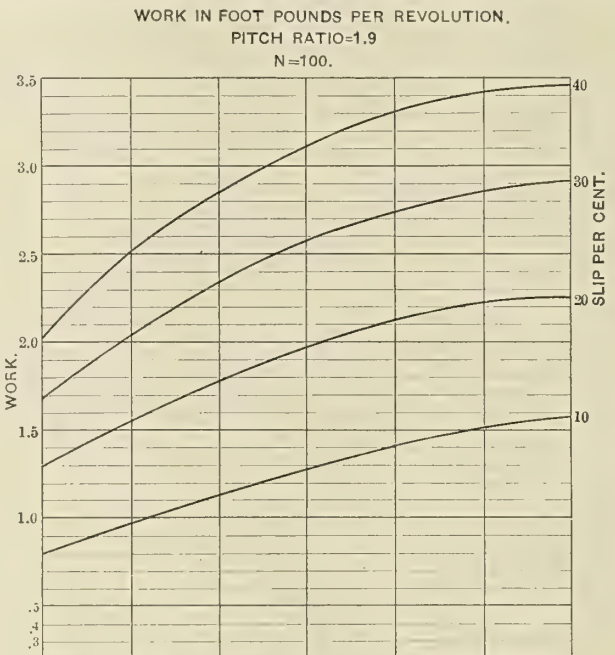


FIGURE 11.

area. For the lowest pitch ratio, however, the decrease of the efficiency with area for these intermediate values of the slip is still well marked. It appears, therefore, that if values of the pitch ratio below 1 are employed, the propeller should be so

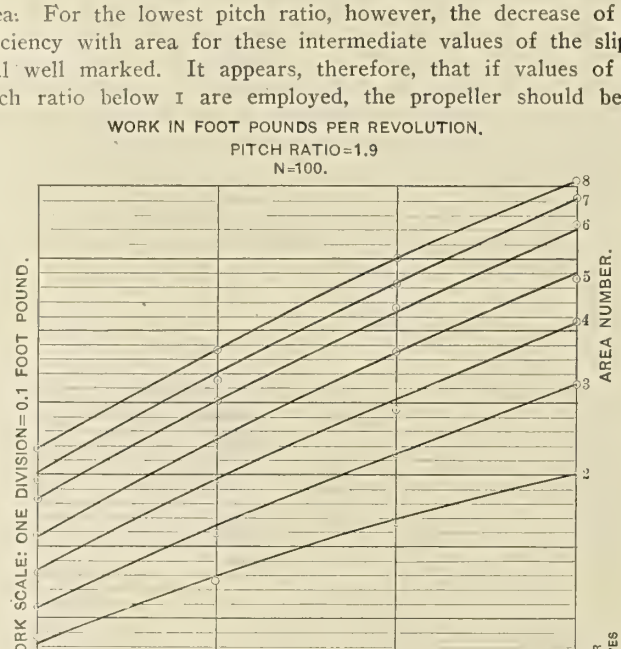


FIGURE 12.

proportioned as to work with a small area and high slip. On the other hand, for propellers lying within the more common ranges of pitch ratio and area, and operating at values of the slip between 20 and 30 percent, it appears that the efficiency will vary but slightly over the entire mid-field range thus indicated, and that any proportions corresponding to operation within these general limits may be freely chosen without fear of any

important drop in the efficiency below the upper limits of 68 to 70 percent.

The application of these results to the problems of design involves the deduction of a series of factors or "constants" relating them in a convenient manner to the indicated horsepower and speed. This operation presents nothing unusual or novel and will not be included in the scope of the present paper, the purpose of which is purposely confined to an actual report of the data taken expressed directly in terms of thrust and foot pounds work per revolution as related to slip and area ratio.

The deduction of such constants as may serve to aid in the operation of design will find a place in the more complete report which will be later published under the auspices of the Carnegie Institution.

THE CRUISER NATAL, ENTERING THE WATER.

The New English Cruiser Natal.

The first warship that the Colony of Natal agreed to make as contribution in the maintenance of the first line of defense has been recently launched. The main dimensions of this vessel, to which the name of *Natal* has been given, are as follows: Length between perpendiculars, 480 feet; breadth extreme, 73 feet 6 inches; displacement at 27 feet draught, 13,550 tons. The normal coal supply carried is to be 1,000 tons, but bunker capacity has been provided for a much increased supply.

The cylinders of the propelling machinery are supported on an "A" frame at the rear, with forged steel columns in front, and are arranged on the Yarrow-Schlick-Tweedy system of balancing, being placed in couples close together, with the valves to the outside. The cylinders, four in number, are arranged for triple compounding. The diameter of the high-pressure cylinder is 43½ inches; intermediate, 69 inches, and of each low pressure 77 inches. Slide valves are used for the two low-pressure cylinders, and piston valves for both high and intermediate cylinders. The stroke is 42 inches, and with the machinery running at 135 revolutions, a total of 23,500 indicated horsepower is expected to be realized. The two propellers are of manganese bronze, with adjustable blades, and work outward when the vessel is going ahead. The main condensers, of gun metal, are oval in section, with brass tubes, there being two to each set of engines, making four in all, each with its own centrifugal pump of gun-metal, with independent engine. The air-pumps are entirely separate.

The *Natal* is fitted with twenty Yarrow boilers and six cylindrical boilers, two-thirds of the power being represented by the former. The proportion of power to weight of machinery is about 10½ horsepower per ton. The speed anticipated is 22.5 knots, and the complement of officers and men is about 700.

The *Natal* mounts six 9.2-inch and four 7.5-inch guns, and her equipment for dealing with torpedo or submarine attack includes 23 quick-firing guns, two 12-pounders and two Maxim guns.

The *Natal* belongs to the first cruiser class in England in which the 6-inch guns have been entirely dispensed with, and although this has involved a reduction in the number of guns, the gain is enormous in the penetrating power of each shot fired. In fact, the 6-inch gun at its best has a muzzle energy of 6,000 foot-tons, while the 7.5-inch gun, 50 calibers in length, develops a muzzle

energy of over 11,500 foot-tons. The 9.2-inch guns, 50 calibers, develop a muzzle energy of 24,000 foot-tons. Assuming, therefore, favorable conditions, so far as rapidity of firing is concerned, the *Natal* will develop 835,000 foot-tons of muzzle energy for each minute of fire, as compared with 829,000 foot-tons in the preceding cruisers. She may fire per minute 48 shots, which can be effective at three miles range.

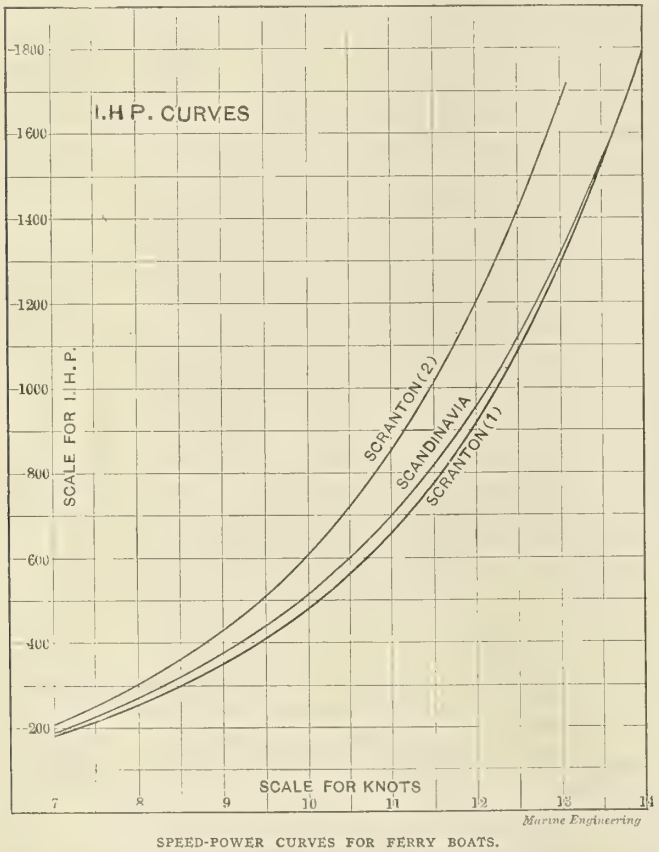
Another important improvement in connection with the armament of the ship is the great height of the gun platform above the sea level. The 7.5-inch pieces are placed within casemates on the upper deck, one on each side, near the center of the length, the protecting armor being 6 inches in thickness. In turrets forward and aft are located two 9.2-inch guns, encased in barbettes of 6-inch armor, decreasing to 3 inches at the base. On the four quarters of the citadel there are similar 9.2-inch weapons, close to the broadside, and as the forecastle structure is cut away, these guns fire in line with the keel, and have an arc of fire of 180 degrees. The side armor is 6 inches thick. This thickness is thinned down by two stages, first to 4 inches, and at the ends to 3 inches.

It has been said that the *Natal* is perhaps the last armored cruiser type to be built for the English navy, more powerful ships being now called for. E. O.

SOME PROBLEMS IN FERRYBOAT PROPULSION.*

BY COL. E. A. STEVENS.

The propulsion of double-ended screw boats offers some difficult problems in screw design. The subdivision of the power shown by the indicator is very complex, and its exact distribution is, with the knowledge at hand, well nigh impossible. There are also few instances in which screw design is of more importance.

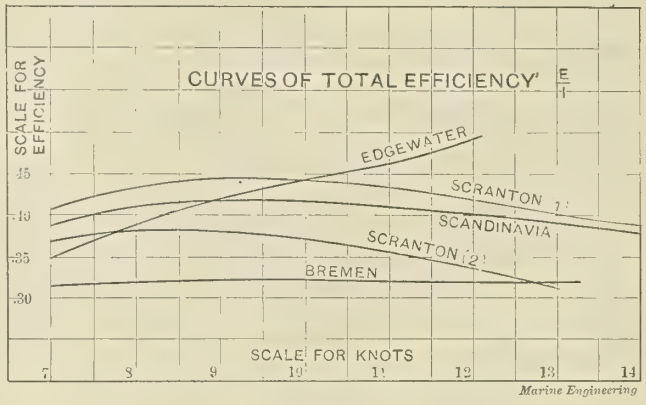


SPEED-POWER CURVES FOR FERRY BOATS.

The *Scranton* class of boats recently put into service on the Hoboken ferries can serve as an illustration. The wheel originally designed for these boats did not give satisfactory results in backing. It was decided to try a wheel of another type. The main dimensions were as follows:

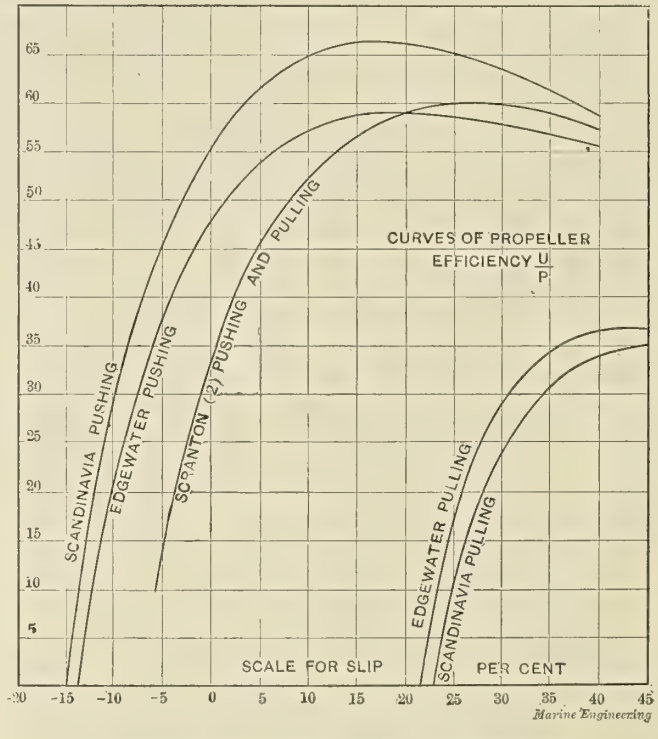
	Wheel No. 1.	*Wheel No. 2.*
Diameter	10′ 0″	9′ 3″
Pitch	11′ 1½″	10′ 6″
Developed area	38.8 sq. ft.	38.5 sq. ft.

The first wheel had blades of oval outline with rounded backs and a directrix curving to the rear. The second wheel was of the so-called DeLamater type, with sections designed to give



TOTAL PROPULSIVE EFFICIENCY.

equal efficiencies in both motions and with vertical directrix. With greatly reduced slip the second wheel consumed about 20 percent more power at the designed speed without any appreciable gain in backing power. On the subsequent boats the original wheel was cut down in diameter and area. The power curves show the results obtained by the *Scranton* with both wheels, and by the *Scandinavia*, a sister ship, with a smaller wheel of the same type.



PROPELLER EFFICIENCIES, FORWARD AND REARWARD.

In discussing these results I adopt the following symbols:

I = indicated horsepower.
F = horsepower consumed in engine friction.
Pl_s = horsepower consumed in stern propeller losses.
Pl_b = horsepower consumed in bow propeller losses.
$Td\text{-}Wg$ = horsepower consumed by difference between thrust deduction and wake gain.

*Read at the thirteenth general meeting of the Society of Naval Architects and Marine Engineers, November 16, 1905.

E_r = horsepower consumed by action of race of forward screw.

E = tow line horsepower.

$P = I - F = P_s$ (power delivered to after wheel) + P_b (power delivered to bow wheel).

Now $I = F + Pl_s + Pl_b + Td\text{-}Wg + E_r + E$.

T can be very closely approximated from tank trials.

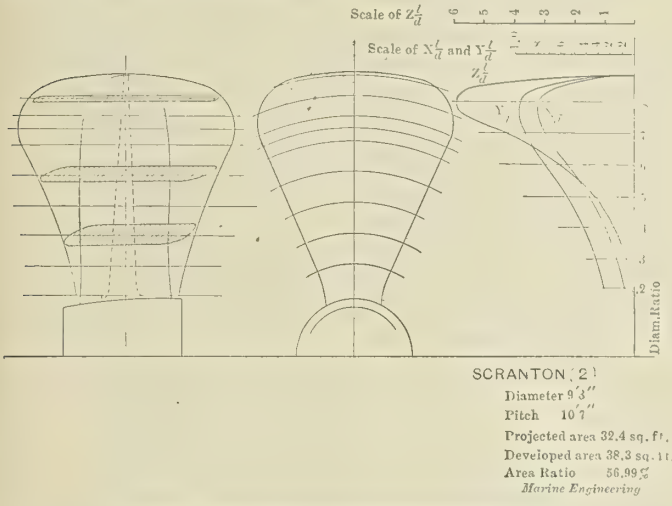
F can be estimated with some approach to accuracy.

$Td\text{-}Wg$ in these boats is probably negligible in design.

Whence $P - E = Pl_s + Pl_b + E_r$.

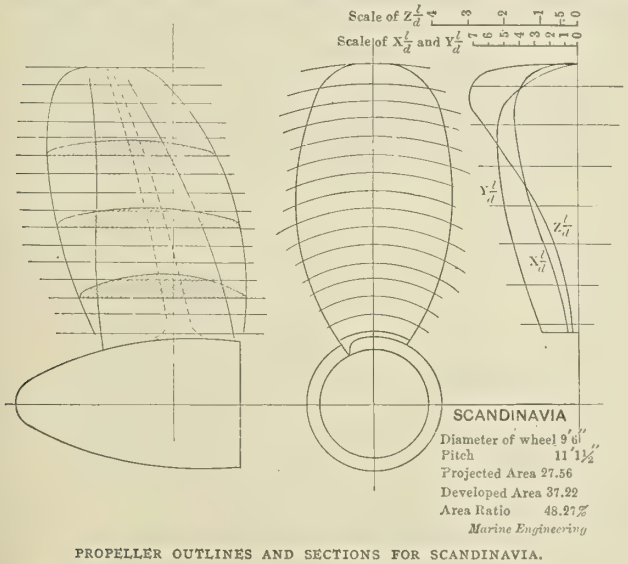
This latter quantity is the one that it is specially desirable to determine.

With a view of so doing, I had model wheels of *Scranton* (2), *Scandinavia*, and *Edgewater* (see Transactions N. A. and M. E., 1902), tested in the United States experimental basin. The resulting curves give the efficiencies and power-absorbing factors for the wheels at various slips.



PROPELLER OUTLINES AND SECTIONS FOR SCRANTON.

Taylor's formula, in which these factors are to be used, is given in the Transactions of 1904. It is, P = constant × Sd^2v^3, in which S is the factor. This formula gives, in this case, much more power than was found on trial. The difference in material and surface, and the fact that the model was accurately finished to pitch, would account for some of the excess power given by



PROPELLER OUTLINES AND SECTIONS FOR SCANDINAVIA.

the model screws. After making reasonable allowance for these differences, some other factor will probably have to be introduced in applying model results to actual practice. This conclusion, I understand, is also shared by Naval Constructor Taylor. In view of this uncertainty, it does not seem advisable to publish these power factors at this time. As before said, we know the values of P and E in this case, but in order to know the values of Pl_s and Pl_b, we must be able to subdivide P into P_s and P_b. To do this from experimental tank data, the wake, and hence true slips, of both screws must be known.

The race from forward screw undoubtedly seriously affects the wake at stern; there is no experimental data as to the wake of the forward screw. With the further uncertainty as to the power absorbed by each screw, it is very doubtful whether any set of assumptions would yield satisfactory results in analysis. The most probable assumptions indicate that the results will not vary much from those given in the analysis of the *Bremen's* trial (see Transactions N. A. and M. E., 1903) in the case of wheels of similar design, such as were used on *Scranton* (2). Such assumptions also lead to a power subdivision and analysis that give explanations of the actual results, as shown by the two trials of the *Scranton;* but they are not so exhaustive as to warrant publication. I submit, however, plates showing curves of total efficiency of propulsion; propeller efficiencies in both motions; and developed outlines, sections, etc., of propeller blades.

As to these curves, I would draw attention to the remarkable difference in total efficiency curve in the case of the *Edgewater* and the other boats. The propeller efficiencies are about what one would expect for wheels of fine pitch, large area ratio, and coarse blade section. They approximate closely to the values given by Taylor's formula,

$$e = (1 - s)\,\frac{asX_c - fY_c}{asX_c + fZ_c},$$

if a be taken at from 4 to 6. I am convinced that the efficiencies are, practically, correct in these cases. The efficiencies in backing are affected by the blade section and form of generatrix.

It should be remembered that the *Edgewater's* sections are coarser than *Scandinavia's*. In view of the uncertainty as to true slips, it may not be out of place to call attention to the fact that the substitution of wheels of the type used on *Scandinavia* may, while reducing power at a given speed, actually increase the maximum strains, both in torsion and thrust on the shafting.

A WOODEN PASSENGER STEAMER FOR ONEIDA LAKE.

From the designs of Mr. Henry J. Gielow, of New York, a new passenger steamer has just been built for the Bay Road Construction Company for use on Oneida Lake. The dimensions include a length of 98 feet overall and 88 feet 9 inches on the waterline, an extreme beam over guards of 22 feet 3 inches, a 7-foot depth of hold and a draft of 5 feet 6 inches. The keel is of white oak 10 by 8 inches, the stem of white oak 8 inches thick and of suitable shapes for maintaining the form desired, with deadwood forward of the same material and thickness. The stern post is also white oak, sided 10 inches at keel and cross seam and 16 inches at center of shaft, and molded 12 inches; the aft deadwood, 10 inches thick, is of yellow pine, with a stuffing box log of the same material 16 inches square. The rudder post is of wrought iron 1¼ by 5 inches, with T-shaped head and feet, and a shoe of wrought iron 1½ by 6 inches extending 6 inches forward of after side of stern post, with two pairs of jaws. The rudder stock is of wrought iron 3½ inches in diameter, tapering to 2¼ inches at the keel, and the rudder frame has a thickness tapering from 2½ inches at the stern post to 1½ inches at the after side. This frame is covered with ⅜-inch wrought iron, the open spaces being filled with white pine.

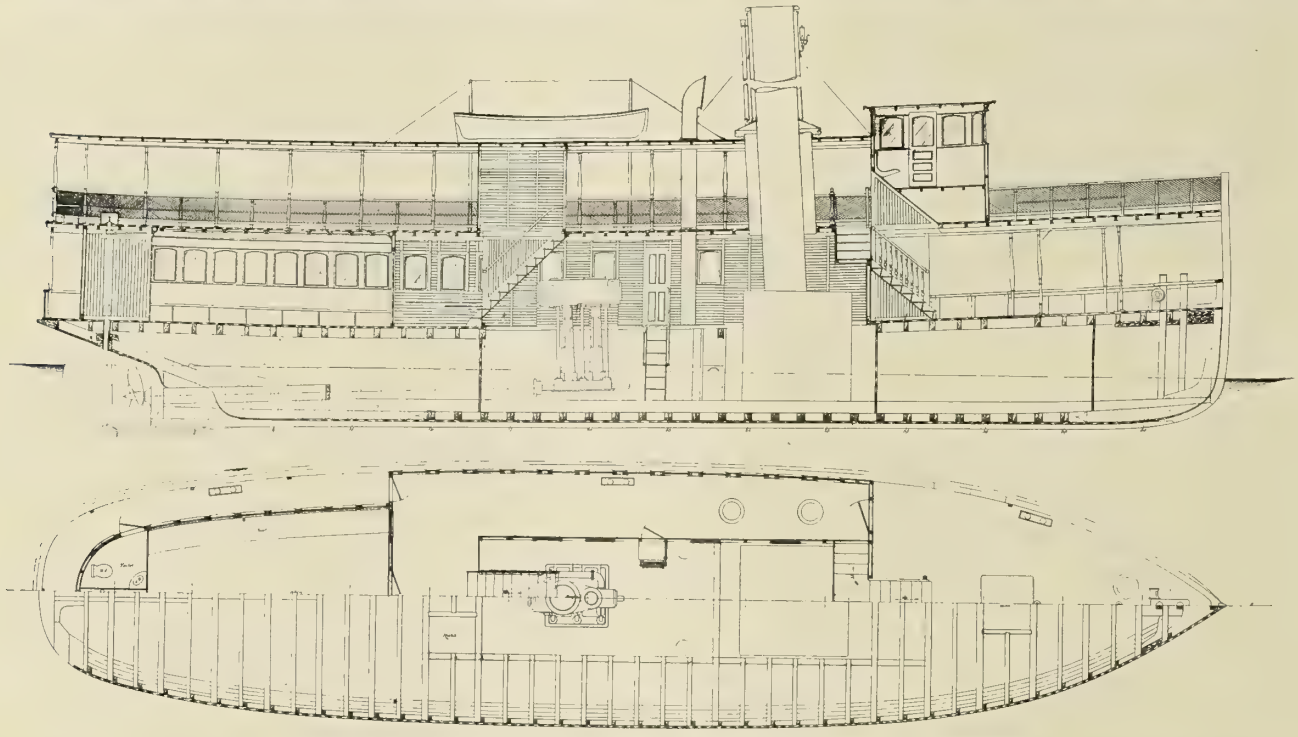
The frames are of white oak, hackmatack and chestnut, spaced 26 inches centers, sided 4 inches and molded 8 inches at keel, 6 inches at turn of bilge and 5 inches at plank sheer. The floors are natural crooks with alternate long and short arms, with 1½ by 2½-inch limber holes on each side of the keel. Under the engine and boiler spaces is a solid filling of yellow pine. The keelson is of pitch pine 10 inches by 10 inches. The garboard

strake is 3 by 10 inches; sheer strake 2½ inches by 9 inches; three wales 2½ by 6 inches, all of white oak. The balance of the planking is of yellow pine 2½ inches in thickness. Wearing pieces are fitted of white oak 6 by 6 inches, with rounded corners, with a pair of fenders extending 12 feet aft from stem. The ceiling is of yellow pine 2 inches thick from clamps to bilge strakes and 1½ inches below that. The bilge keelsons are of yellow pine 6 by 8 inches; beams of the same material molded 10 inches at centers and 5 inches at ends and worked with a crown of 5 inches amidships; siding 4 inches and 6 inches respectively.

The plank sheer is of white oak 2 inches thick and 12 inches wide, let down over the stanchions; the deck planks are dry, white pine 2 inches by 3 inches; the rail is of white oak 2¼ by 6 inches with top rounded off; rail 1⅛ by 4½ inches; bulwarks are of white pine 3 by ⅞ inches.

There are three bulkheads, located respectively at frames 16, 31 and 39, being of yellow pine 2 inches thick and stiffened by intermediate stanchions 3 by 4 inches. Coal bunkers are located on each side of the boiler, and constructed of ¼-inch steel plates and 10 feet wide, with forward angles rounded and front rounded to a radius of 9 feet 8 inches; with clear head room of 6 feet 6 inches. Carlines are 1⅞ by 2⅞ inches and deck ¾ by 3 inches, covered with cotton duck. Sills are of white pine 5 by 5 inches rabbeted for receiving sheathing, which is of white pine 1 inch thick outside and ⅞-inch in ceiling; studs are 2 by 3 inches of spruce. The steering wheel has a diameter of 6 feet over spokes, and is made of locust and mahogany, with suitable equipment of chocks, blocks, sheaves, chains, etc., for serving its purpose. A spirit compass with binnacle and lamps is fitted on a shelf forward of the wheel.

From the after end of pilot house to the stern of the vessel, a distance of 71 feet, and extending the full width of the saloon deck, is a space known as the awning deck, with head room 6 feet 10 inches at center and 6 feet 8 inches at sides. The deck is ¾ by 3 inches of white pine, covered with No. 6 cotton duck. The outboard ends of carlines are supported by twenty-five turned spruce stanchions 2 inches in diameter, and eight galvanized wrought iron stanchions of the same diameter. Outside these awning stanchions and 3 feet above the deck is a white



INBOARD PROFILE, HALF DECK PLAN AND HALF HOLD PLAN OF ONEIDA LAKE STEAMER.

stiffened with 2 by 2½ by ¼-inch vertical angle bars spaced 21 inches. Coamings for engine and boiler hatches are of yellow pine 5 by 6 inches.

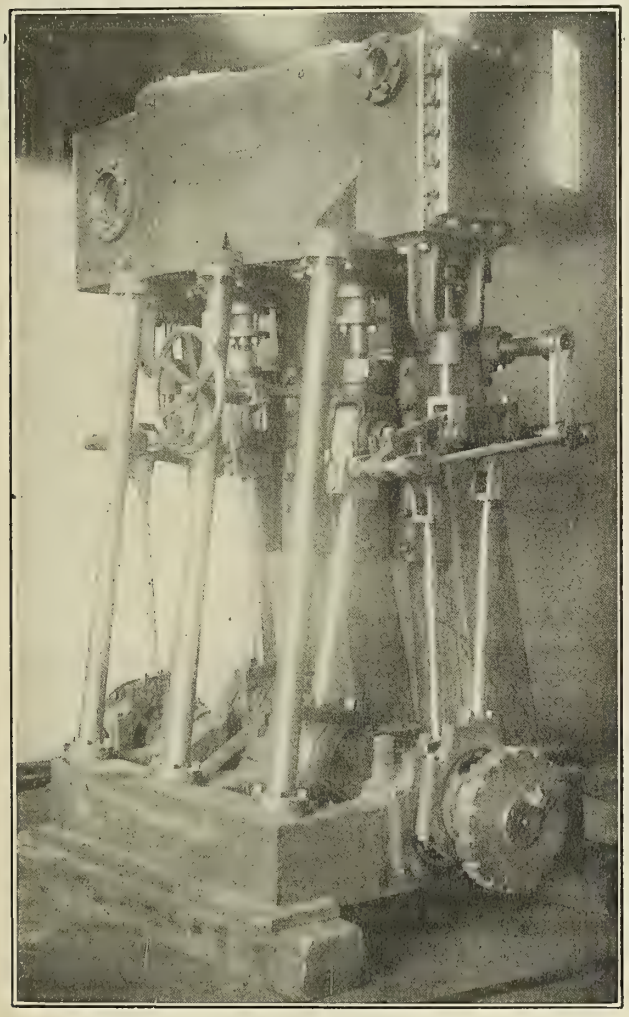
The main deck house has a length of 65 feet 6 inches, the sides forward for a distance of 39 feet 9 inches running along the inside of the rail, while the after end allows a clear passageway inside the rail of 2 feet 6 inches on each side. The head room under the carlines is 7 feet 4 inches. The studs are of spruce 4 by 3 inches, the clamp of yellow pine 2 by 5 inches extending the full length of the hull, carlines of the same material 2¼ by 3¾ inches, 21 inches apart and extending from side to side in one length. The deck extends the full length and width of the vessel, and is of white pine 1 by 3½ inches, covered with No. 5 cotton duck and heavily painted. The outside sheathing is of white pine 1 by 3½ inches, and ceiling of the same material ⅞ by 3½ inches laid horizontally.

The center deck house, inclosing the engine and boiler spaces, is 32 feet long, 10 feet wide, with plates 3 by 5 inches, sills 5 by 6 inches, studs 2 by 3 inches and corner posts 4 by 4 inches, rabbeted to receive sheathing, which is of white pine ⅞ by 3 inches. The pilot house is on top of main deck house, 9 feet 8 inches long pine rail 2 inches square and supported by intermediate turned spruce stanchions 1¾ inches in diameter.

There is an inverted surface condensing compound engine, built by the Morris Machine Works, Baldwinsville, N. Y., with two cylinders and cranks at an angle of 90 degrees, operating a single screw. The high-pressure cylinder has a diameter of 14 inches and low-pressure 28 inches, both with 18-inch stroke. The steam ports of high-pressure cylinder measure 13⅜ and exhaust ports 31⅝ square inches. The low-pressure cylinder has steam ports measuring 38½ and exhaust port area 60½ square inches. The cylinder barrels have a thickness of 1⅛ inches, with counterbore of 3/32 inch in each end. Both cylinders and steam chests are to be lagged with felt and Russia iron. The steam chests are cast in one piece with the cylinders, and a relief valve is fitted on each. The slide valves are of the ordinary D form, of cast iron. The bedplate of the engine is one casting, composed of two fore-and-aft and three transverse box girders, the framing consisting at the back of the engine of two cast iron box columns measuring outside 12½ by 13 inches at the bottom and 8 by 9 inches at top, with shell one inch thick, and with crosshead guides 7 inches wide cast on inside faces.

The front columns are wrought iron, 3½ inches in diameter, finished, and fitted with suitable flanges for fastening to cylinders and bedplates.

The pistons are of cast iron double shell, with cast iron followers and packing rings. The piston rods are of mild steel, 2⅝ inches in diameter, screwed into pistons and crossheads. The latter are of cast steel with journal 4 inches diameter and 5 inches long. The crosshead slippers have a wearing surface of 84 square inches. The connecting rods are of hammered iron 3 feet 10 inches long between centers and measuring 2¾ inches in diameter of neck at crosshead end, and 2¾ by 4 inches at crank end. The crank shaft is of forged steel in one piece with crank arms 4½ inches thick and 7¾ inches wide. Crank



THE COMPOUND ENGINE.

pin journals are 6½ inches long and 6¼ inches in diameter, and the line shafting is fitted in two lengths, of forged steel, 6 inches in diameter with cast steel disk coupling. The bearing surfaces on thrust bearing receive a maximum pressure of not more than 40 pounds per square inch. Besides the thrust bearing there are three main bearings 6¼ inches in diameter, and respectively 8, 11¾ and 8 inches long. All are fitted with removable babbitt-lined shells. The valves are operated by the Stephenson link motion, of the single bar pattern, with a throw of about 12 inches. Each link is operated from two eccentrics with 1¾-inch face. The links and valve motion are made of steel, with take-up on all connections. The valve stems are $1\frac{7}{16}$ inches in diameter.

The air pump and jet condenser is of a horizontal double-acting type, with steam and air cylinders respectively 8 and 14 inches in diameter and 12-inch stroke. The condenser is of the cone type, fitted with composition spray valve. The feed pump and injector are of the horizontal duplex type, with steam and water cylinders 5¼ and 3½ inches in diameter with a 5-inch stroke, and with water end designed for a working pressure of 200 pounds per square inch and a temperature of 150 degrees. A bilge ejector with a discharge nozzle 2 inches in diameter is fitted. The main and exhaust steam pipes are of pressed copper, 4½ inches and 8 inches in diameter respectively. The electric light plant consists of a direct-coupled generating set with an output of 7 kilowatts at 550 revolutions per minute. The engine is vertical, 5 by 4½ inches, and fitted with an automatic governor; the voltage is 110. There are a searchlight 13 inches in diameter and 60 incandescent lights.

The propeller, which has four blades, is of cast iron, 5 feet 3 inches in diameter and with a pitch of 8 feet 6 inches, giving a pitch ratio of 1.62; the projected area of the four blades aggregates 1,620 square inches. The stern bearing is 15 inches in length, fitted with lignum vitæ staves, and with a flange against the aft side of stern post. The inboard stuffing box is of cast iron with a shaft bearing of 12 inches; the usual gland is fitted to the shaft and holds the packing in place.

Steam is supplied by a safety water-tube boiler 9 feet long, 9 feet wide, 6 feet 5 inches high and tested to 500 pounds hydrostatic and 250 pounds steam pressure. It has a grate surface of 57.6 square feet and a heating surface of 1,328 square feet, giving a ratio of 23.1 to 1. The steam drum is of mild Siemens-Martin steel of a tensile strength of 60,000 pounds per square inch, and of such ductility as to give a reduction of area of 50 percent. The grate bars are single, with cast iron bearers. The furnace has two fire doors. The boiler is jacketed and lagged with magnesia. The boiler keelson consists of 2 I beams 10 feet long, 6 inches high, with a web ¼ inch thick. The single funnel is constructed of No. 12 B. W. G. steel with an inside diameter of 42 inches and a length of 30 feet. The outside stack is of No. 14 B. W. G. steel 50 inches in diameter.

Stranding of the Steamer Alicia.

Our photograph, which was sent us by Chief Engineer George H. Waring, Jr., of the steamship *Captain A. F. Lucas*, shows the position of the Spanish steamer *Alicia*, when ashore in May, on Ajax Reef, near the Fowey Rocks Lightship, Florida. The steamer was eventually taken off, after having been on the

THE ALICIA AGROUND OFF THE FLORIDA COAST.

reef about a month. The *Alicia* is described as an iron screw vessel with a length of 345 feet 2 inches, a beam of 37 feet 4 inches, and a depth of 25 feet 9 inches. She is divided by five bulkheads into six watertight compartments, and has a registered tonnage of 1,768 net and 2,795 gross. She is propelled by one compound engine with cylinders 38 and 71 inches respectively in diameter and a 48-inch stroke, and was launched at Glasgow in October, 1883.

MOTOR BOATS, VIII.

BY DR. WILLIAM F. DURAND.

THE DESIGN OF FORM.

The design of a motor boat as a whole is readily seen to divide itself into the two main problems of the design of form and the design of structure. In the present chapter we shall consider the design of form. In the preceding chapter we have discussed the principal characteristics of form and their relations with the various qualities of the boat, such as speed, safety, carrying capacity, etc. We have now to consider methods for determining in a regular and systematic manner such general form characteristics as may be desired, and for the representation of such form in manner best suited to the needs of the practical builder.

Before taking up the discussion of actual designing methods, however, brief reference must be made to certain so-called form or fineness coefficients.

Let the length of the boat on the water line be denoted by L, the breadth by B and the draft by H. Then the product B × H will be the area of a rectangle which will just contain the immersed midship section. Let M be the area of the actual section.

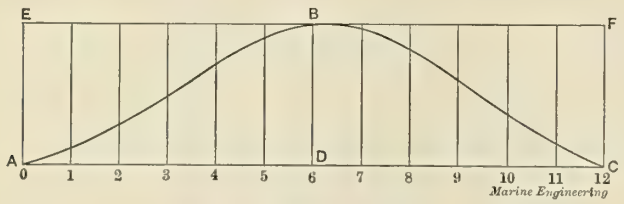


FIG. 39. CURVE OF SECTIONS.

Then the ratio M ÷ BH is called the *midship section coefficient* and represents the fraction of the total area which the actual section contains.

Likewise the product L × B will be the area of a rectangle which would just contain the water plane or horizontal section at the water line. Let A be the area of the actual water plane. Then the ratio A ÷ L B is called the *water plane coefficient* and represents the fraction of the total rectangle which the actual water plane contains.

Again the product L × B × H will be the volume of a box which would just contain the under water form. Let V be the actual volume of the under water body. Then the ratio V ÷ L B H is called the *block coefficient,* and evidently represents the fraction of the total box volume which the actual boat form contains.

Again the product L × M will be the volume of a solid of section constant and equal to M, and of length equal to that of the boat. Then the ratio V ÷ L M is called the *prismatic coefficient,* and represents the fraction of this total volume which the actual boat form contains.

It is readily seen that the indications of the midship section coefficient refer solely to the form of this section, and give an estimate of its fullness compared with a rectangle as the standard. The indications of the water plane coefficient, similarly, refer solely to the form of the load water line, and give an estimate of its fullness compared with a rectangle as the standard. The indications of the block coefficient, on the other hand, refer to volume, and give an estimate of the fullness of underwater body compared with a rectangular block or prism as the standard. Similarly the indications of the prismatic coefficient refer to volume and give an estimate of the fullness of underwater body compared with a body having length L and a constant cross section M. A little thought will also show that this coefficient refers solely to what may be termed fore and aft fining.

Suppose again the areas of a series of sections taken to the load water line, and let them be denoted by M_0, M_1, M_2, etc. Let these be plotted on an axis of length each at its own station. The result will be a curve similar to that of Fig. 39, and called a curve of sections. The maximum height of the curve, *B D*, represents, of course, the area of the midship section, or M, as noted above. The distance *A C* represents likewise the length L. Then the product L M is represented by the area of the rectangle *1 C F E*. The area of the curve *A B C D A* in a similar manner represents the volume of the actual boat form, which we have denoted by V.

The complete proof of this last statement would require an application of the integral calculus, which would take us outside our purpose in the present series of articles. The fact may be seen in a semi-mathematical manner as follows:

It is clear that the volume of the boat will be measured by the length multiplied by the mean or average cross sectional area. It is likewise clear that the area *A B C D A* will be measured by the length *A C* multiplied by the average vertical dimension or ordinate. But the length *A C* represents the length of the boat, and the ordinates of the curve represent the cross sectional areas. Hence the product of the length *A C* by the average ordinate will be proportional to the product of the length of the boat by the average area of section. Hence the area of the curve will be proportional to the volume of the boat, V. It is then clear that the ratio V ÷ L M will be the same as the ratio between the two areas *A B C D A* ÷ *A C F E*. It follows that the prismatic coefficient is the area coefficient of the curve of sections, *A B C*, and shows the fullness of the area *A B C D A* compared with the rectangle *A C F E* as standard.

The usual range of value in the midship section coefficient is from .65 or .70 in the peg-top form of section as in Fig. 40, to .90 or above in the flat floor and sharp bilge form as in Fig. 41.

The usual range of values in the water plane coefficient is from .65 or .70 in very fine forms of the old or ship type with deadwood, through .75 or .80 for fine forms with torpedo boat or flat stern, and to .85 and upward for very full forms.

The usual range of value in the block coefficient is from about .40 or .45 for fine forms to .50 or .60 for moderate and to .70 for full forms. Extreme forms for high speed may have a coefficient as low as .35, and the whole range of boats for pleasure and high to moderate speed may be expected to show values of the block coefficient from this figure up to about .50.

The usual range of value in the prismatic coefficient is from about .6 for very fine high speed forms to .70 or .75 for moderate speeds, and to .80 or .85 for low speed and large carrying capacity.

We may turn now to the various plans and lines by means of which the form of a boat may be represented on the drawing board.

If we suppose a series of planes passed parallel to the surface of the water they will cut out from the form of the boat a series of lines known as water lines and having the general form shown in Fig. 36 (lower plan). Since the two sides of the boat are the same it is customary to show the lines for only one side. Hence the plan showing such lines is known as the *half breadth plan.*

If, again, we suppose the form cut by a series of planes vertical and transverse, or perpendicular to the base line, they will cut from the surface of the boat a series of lines known as *sections* and having the general form shown in Figs. 36, 34 (small plan). Here also, since it is unnecessary to show more than one side of the boat, it is customary to show on one side the sections of the fore body and on the other that of the after body. Such a plan is known as a *body plan.*

The plan shown in Fig. 36 (upper plan), and which gives the side elevation of the form, is known as the *sheer plan.* If the boat is cut by a series of fore and aft vertical planes, the lines cut out from the surface are shown in such a plan, and are called *bow and buttock lines.* Referring again to Fig. 34, if

the form of the boat is cut by a series of oblique planes running fore and aft and lying as nearly as possible perpendicular to the surface at various depths, they will cut from the surface a series of lines which may be shown, each in its own plane. Such planes, indicated by *A, B, C, D, E,* Fig. 34, are known as diagonals, and the resulting lines are shown in developed or true form in the upper plan on the right. The chief use of such lines is in fairing up the other plans so as to insure a smooth and continuous form, especially near the bow and stern, where the indications of the other plans are sometimes less definite or exact.

Referring now to Figs. 36 and 34, the water lines may be fined away sharply fore and aft, beginning from the midship section; or they may carry nearly the full width for some little distance fore and aft and then fall away more abruptly to the pointed extremities. Such forms of water line are more specially typical of the ship form with deadwood. On the other hand, with the torpedo boat stern the typical form of water plane is shown in Fig. 36. It is sharp pointed at the bow, and gradually widens, gaining its greatest width frequently at some little distance aft of the mid length, and then rounding off to the stern as shown.

With the ship form of under water body the fineness of water plane will increase as the required speed is higher, while as speed requirements are less exacting and more carrying capacity is required, the water lines will become fuller in form, and the bow and stern will become less sharply pointed. With very full boats of this form the after body should preferably be somewhat finer than the fore body, in order to diminish eddy resistance at the stern and facilitate the flow of water to the

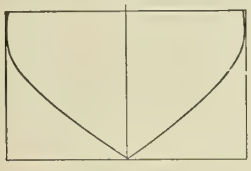


FIG. 40.

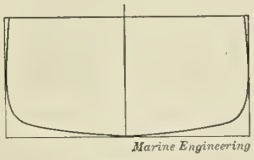


FIG. 41.

propeller. With the torpedo boat type of stern the form of the water lines in the after body shows but slight variation, with general fullness of form. With the fore body, however, the general fineness of water line is greater, and the angle of entrance is less as speed requirements are greater.

Turning now to the body plan, the sections may at one extreme be nearly rectangular, or at the other extreme more nearly triangular, or even hollowed in as near the stern in ship-formed bodies. The general character of the transverse sections in any one boat is to a considerable extent determined by the form of the principal or midship section, which thus becomes a general index to the form so far as it may be determined by the sections. The usual form of midship section for motor boats shows a moderate to well marked rise of floor, as in Figs. 36 and 34, with a well-rounded bilge and side, often sloping outward slightly at the water line and rising to the vertical near the rail. Within this general type the variations in form of midship section are determined not so much by requirements of speed as by minor or secondary considerations. Thus, if especially shallow draft is desired the midship section will have a rather flat floor, with relatively less rounding off on the bilge, and consequently a high midship section coefficient. On the other hand, if deep draft on moderate displacement is desired, the form of midship section will tend toward the "peg-top" type, that is, with sharp rise of floor and well rounded bilge. In general, however, the flat, shallow section is not the most favorable to seaworthiness, nor is the relatively sharp bilge of Fig. 41 as favorable to low resistance as the more rounded form of Fig. 34. For the best combination of speed, seaworthiness and generally desirable qualities, both extremes are to be avoided, and the prevailing form of midship section in such boats is usually found somewhat as indicated in Figs. 36 and 34.

With the torpedo boat type of stern the after sections become shallow as the stern is approached, and relatively full compared with the surrounding rectangle. This, however, does not necessarily denote fullness of form as determined by cross sectional area compared with that of the midship section. It simply means, as already explained, that the fining away is vertical rather than horizontal.

Toward the bow, with both forms of stern, the sections approach the V or U shape, according to the designer's choice. The latter are presumably somewhat more favorable with regard to resistance, while the former, especially above the load water line, will tend to give lifting power to the bow in a seaway, and hence to reduce the pitching and amount of water shipped when running into a head sea.

Having thus discussed the various geometrical plans and projections and coefficients of form, we may undertake consideration of the problem of actual design. It should first be noted that but rarely is such a design absolutely new and without a more or less direct dependence on previous forms. The accumulation of such forms in any drawing office furnishes a series of starting points from which, by slight modifications in various ways, a vast number of derivative forms may be produced, and in this manner the conditions for new problems may be readily fulfilled. Such modifications may be of two kinds: those involving a change in the proportions of length, beam and draft or depth, and those involving a change in some or all of the fineness coefficients as discussed above.

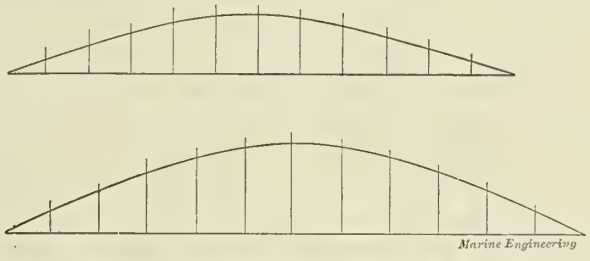


FIG. 42. EXPANSION OF HALF-BREADTH PLAN.

In the simplest possible case it may result that the proportions and form coefficients may all be considered satisfactory. The only change in such case is one of dimension or scale. No new design is therefore required, and the conditions may be realized by a direct increase or decrease of all dimensions in the same ratio. In such case it is not as a rule necessary to make any change on the drawing board or to redraw the lines to the new scale. All such changes may be made when the boat is laid down in the boat shed, and in general manner as will be described at a later point.

A more common case arises in connection with the acceptance of the various coefficients of form, but with requirements which call for some change in the proportions of the three principal dimensions L, B, and H. Thus let us suppose that a given set of lines relates to a boat 24 feet long, 5 feet beam, and 2 feet draft. It is desired to use the same characteristics of form for a boat 30 feet long, 5 feet 3 inches beam, and 2 feet 3 inches draft. The various proportions between the new and old would, therefore, be as follows:

Length ratio $= 30 \div 24 = 1.25$
Beam ratio $= 63 \div 60 = 1.05$
Draft ratio $= 27 \div 24 = 1.125$

This means simply that the new boat is to be 25 percent longer than the old, 5 percent wider and 12.5 percent deeper. Here again no work whatever need be done on the drawing board, as the changes required may all be made when laying the form down full size. The general method of procedure with these ends in view may be outlined as follows:

The first step will be to obtain the new water line or half breadth plan. To this end we note that the intervals between the sections must be increased in the same ratio as the length as a whole. Thus suppose in the original drawing that 12 sections were employed, giving a spacing of 2 feet between each, then the new spacing will be similarly 30 ÷ 12 or 2.5 feet. The ordinates at each end of these stations must then be scaled from the original lines or taken from the so-called table of "offsets" and then multiplied by the breadth ratio 1.05. This will give the lengths of ordinate to be set off at the new stations, and the resulting lines will constitute the new half breadth plan as desired. This operation for a single water line is indicated in Fig. 42. The new length of 30 feet may then be divided into any frame spacing as desired and the locations of the frames drawn in. To determine the form of section at these stations, and hence the actual form of the frames, the vertical spacing of the ordinates must be increased in the ratio of the drafts, or 1.125. If therefore the old spacing was 3 inches the new must be 3 × 1.125 or 3.375. Horizontal lines laid down to this spacing may then be taken as the basis of the body plan, and the ordinates, taken from the new half breadth plan, and laid off on these lines, will give correctly the forms of the sections or frames as desired.

In laying down a new design by this method, and after the form has been determined as above, it may be desired, as a matter of record, to lay down a new set of water planes in the body plan, spaced at some even interval, such as 3 inches. The ordinates taken from such a set of planes may then be made a matter of record in a table of offsets. For such purpose it will be well to distinguish the new set of planes by a different color or by some other arbitrary distinction in order to avoid confusion.

In any such problem of design many practical points will arise which cannot be completely treated in the present brief account of general principles. By the intelligent application of such principles, however, and in the general manner indicated, it will be found a simple matter to lay down a new set of lines having the same characteristics as the initial or type form, but with the proportions of L, B, and H modified in any manner desired. It may sometimes happen that one or more of those ratios may remain the same in the new as in the old form. Thus it may be desired to retain the same ratio of L to B or B to H, or indeed to retain all ratios the same, in which case the change becomes one of size alone, already discussed as the first and most simple case. Again in case the new dimensions are smaller than the old, or some smaller and some larger, the same general principles apply. In short, for all cases, no matter how special, where the form coefficients are to remain the same and a new form is to be derived with new dimensions and new proportions of L, B and H, the same general procedure will serve to determine all features of the new form as may be desired.

Engineering and Shipbuilding Features in Scotland.

Some interesting labor developments have occurred during the year in the engineering trade of this district; and they are of national importance because the Federation of Engineering Employers extends over the whole country, and on any question of principle, or method of procedure, the federation is bound to support any district association against the trades unions. Both the patternmakers and what are called here the engineers (but in America machinists), demanded an advance in wages of ½d. an hour. This was declined by the employers as not warranted by the condition of trade, and also because the contracts which are at present keeping the shops engaged were booked on the basis of existing wages. The patternmakers would not listen to reason, and would not wait for a conference, but went on strike. They thought to corner the employers, but as a matter of fact they have been a stimulus to them. The strike of patternmakers on the Clyde promises to establish a system of mutual aid among the employers which has in it possibilities of great development. When a shipbuilder builds more than one vessel of the same size and type there is, of course, a great saving in office work, in patternmaking, and in other departments of the shipyard; and so in the engine shops. Employers usually endeavor to make one set of patterns suffice for more than one vessel or one set of engines. This plan has been confined to individual works, but the patternmakers' strike has shown that it is capable of expansion by a method of borrowing and lending in the matter of patterns. A pattern received from another firm may not always suit an engineer exactly, but it may be so near as, with very little alteration, to serve his purpose. This idea of uniformity and standardization is steadily spreading among shipowners and shipbuilders. Owners frequently order steamers to be built from the same drawings, not at one yard, but at separate yards. In such cases builders do not think of making two sets of patterns, and the strike on the Clyde has done a great deal to further the idea of mutual assistance of this kind among the associated employers. Thus the strike was frustrated by the free interchange of patterns. Members of the federation were prepared to send their patterns anywhere, so long as they were useful to other members, and the result of this policy will be greater uniformity in the future designing of vessels and engines. The system was got into such good working order that any employer requiring a pattern could know within twenty-four hours whether there was such a pattern in the district, and have it immediately transferred to his shop. The idea was to tide over the dispute, but there is no doubt the matter will be put on a permanent business footing. What with the help of exchanged patterns, the work of apprentices, and devices for making simple castings, without any patterns whatever, the inconvenience of the strike was greatly minimized, and the employers concluded that they would not allow the men simply to walk back at leisure to their work. They would have "terms of settlement" somewhat on the lines of those between the federated employers and the machinists. These terms provide first for local, then for central conferences, pending any dispute, and without cessation of work, and on these terms the present demands of the machinists are being dealt with.

Benj. Taylor.

Gasoline Marine Engines.

Editor Marine Engineering:

The present situation as regards gasoline marine engines is peculiar—to put it mildly—although more apparent perhaps to those who make a specialty of marine work than to others, even the majority of marine engineers failing to appreciate it. The statement, however, of one of them to the writer, covering his views on the subject, nearly hits it, and is as follows:

"The gasoline marine business is surely in a bad mix, and it looks as if most of the manufacturers don't know what a boat that is big enough to be good for anything has to contend with, or what a complete outfit of driving machinery should consist of; and as to the *horsepower*, I don't believe that one in ten has any idea of what this means, but rate their engines to compete with some other fellow, or beat him a little, and consequently bring about a lot of trouble by fooling a customer, even though they don't know that they are doing so."

While this expresses the views of some well-posted marine (steam) engineers, and is unfortunately true, to a surprising extent, there are those building the gasoline engines who fully understand every detail pertaining to the make-up of a first-class outfit of marine machinery, and who must find it difficult to reconcile the rated power of their engines as compared with steam, to the *actual* power transmitted to the propeller or paddlewheel.

Even if we eliminate a few hundred makes by cutting out the small motors entirely, and therefore all of the two-cycle engines, we have an assortment of different designs remaining that must be confusing to the intelligent buyer who is seeking for *the best* and is willing to pay a fair price; for the best gasoline marine engine of to-day is not so much a matter of individual opinion as the buyer might be influenced in believing through reading the

various catalogues and listening to the explanations of the salesmen. The engine itself, or rather *the complete outfit of driving machinery,* must stand on its own merits, and every piece and part of each accessory, as well as the engine itself, must be *right,* in order to insure that dependability which the practical user of steam machinery insists on. Above all other considerations it should be proportioned for and suited to the character of the service in which the boat is to be engaged. Most gasoline engines are designed for light high-speed launches turning a small wheel, barely submerged, at a very high speed, and then calculated as exerting a certain horsepower, but when that engine is actually put to work in a fairly stanch hull, and with a consistent size of working wheel properly submerged, there invariably follows a panicky inquiry as to "where has all the power gone?" It is simply a misapplication, and a different type of engine with longer stroke and slower speed should have been used.

Very few have had the opportunity, and availed themselves of it, as the writer has, to keep continually in touch with the commercial value of medium-size gasoline marine engines (complete outfits) as compared with steam, and the feature that has been most noticeable during the past several years is the effort, that would almost seem to have been made purposely by the gasoline-engine builders, to handicap their own product for the benefit of the marine *steam*-engine builders, by outdoing each other through catalogues and letters, as well as verbally, in offering their product as being practically automatic; requiring little or no attention; always ready at a moment's notice; never any trouble; very economical, and claiming about 50 percent more power than the engine could possibly transmit.

Perhaps this is natural in the evolution of a new power and the conditions under which so many thousand gasoline engines are manufactured, but it was as noticeable twelve years ago as it is to-day; and it would seem as though it were time to call a halt on such statements and offer not only a properly-built and well-equipped complete outfit of gasoline marine machinery, suitably designed for the work it must do, and at a fair working profit, but also *inform the buyer* that it *does* require intelligent attention, and should always receive it, or otherwise that trouble may be expected; furthermore, that the engine cylinders are of a certain diameter and certain length of stroke, and that the power it is able to transmit may easily be determined by him, or by anyone sufficiently interested to figure it out, by simply multiplying the square of the diameter of cylinder by the length of stroke and working revolutions per minute, and then dividing by 15,000 for a one-cylinder engine; double the result for a two-cylinder, and so on; for it has been determined through comparative tests—principally electric—that 15,000 is a conservative, and therefore proper, constant to use.

It would seem unnecessary to touch upon the now well-known fact that the ordinary stationary gasoline engines cannot be turned into so-called boat engines by making a few changes on the main machine, any more than the stationary steam engine can be treated in that manner, although a few years ago it was persistently tried, with some profit to the seller—at a severe cost, however, to the user.

The machinery required is of a character that is designed and built to order to meet the requirements of each particular case, and not to offer a light high-speed gasoline outfit as suited to a heavy business boat, or *vice versa,* but to *fit the case* at a fair price; and when that is done, and not until **then,** will the gasoline marine engine be recognized as a practical working success.

A BUILDER OF MARINE STEAM ENGINES.

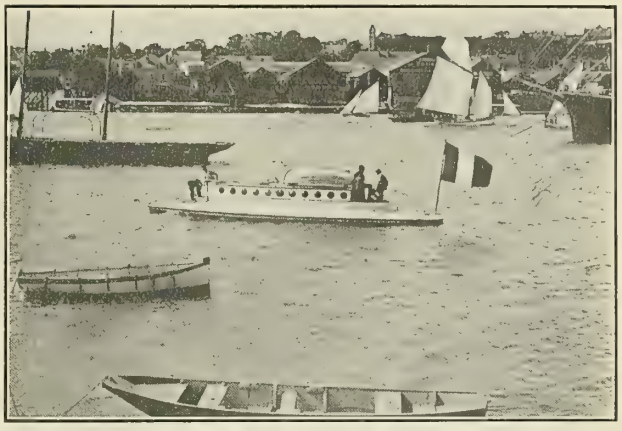
Personal.

W. I. Babcock, late general manager of the Chicago Shipbuilding Company, has been appointed New York representative of the Bath Iron Works, Bath, Maine.

Robert S. Riley has severed his connection with the New York Shipbuilding Company, of Camden, N. J., and become consulting engineer for the Enterprise Transportation Company, at pier 29, East River, New York.

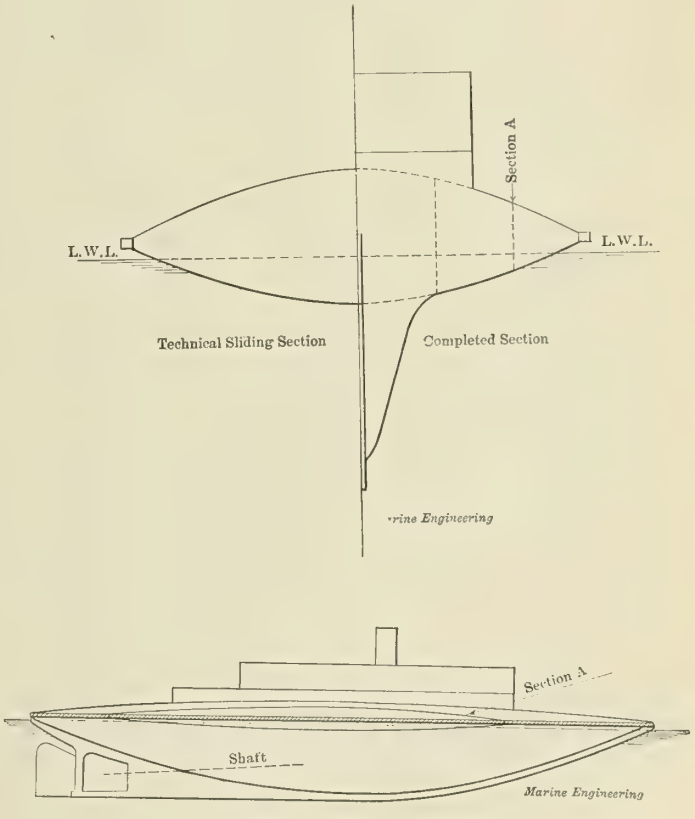
The Motor Boat Slider.

A peculiar model of motor boat, designed to slide over the water, having good stability and a moderate speed, and with an appearance much like a submarine boat, has recently been put into



THE SLIDER UNDER WAY.

service in French waters, and is illustrated in the accompanying cuts. The boat at present in operation has a length of 39 feet 4 inches, a beam of 9 feet 10 inches, and a draft of 5 feet 4 inches, the displacement being about 6 tons. The engines run at 400 revolutions, and with four cylinders develop 66 horsepower, giving a speed of about 13 knots with one form of screw and 9½ knots



SECTIONS AND PROFILE OF SLIDER.

with another. This boat is found to pitch and roll very little under swells and heavy seas, due largely to the peculiar form of the hull. It is built of steel with plates about ⅛-inch thick.

The owner estimates that by building a boat about this same length, with a beam of 13 feet 1 inch, and draft 4 feet 10 inches, and providing it with very powerful machinery, a speed of as high as 32 knots may be obtained.

J. PELTIER.

Marine Engineering

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Changes to be made in copy, or in orders for advertisements, must be in our hands, in New York, not later than the 10th of the month, to insure the carrying out of such instructions in the issue of the month following.

The edition of the February issue of Marine Engineering *comprises 5,500 copies. We have no free list, accept no return copies and issue only enough to supply the regular demand, so that the 5,500 copies represent legitimate circulation.*

The Printers' Strike.

As many of our readers are aware, the printing business in New York City is at present in a very precarious position because of a general strike of job printers inaugurated on the first day of January. We do not desire to enter into the ethics of the question, so far as the demand of the union printers for an eight-hour day with their previous nine hours' pay is concerned, but as we are vitally interested in the subject from the fact that our publishers are seriously handicapped by the strike, we think it best to give our readers some idea as to what is going on, and to ask their indulgence in case it should prove difficult or impossible to issue this number (or the next) exactly on time.

The one salient feature in the proposition submitted to the employers by the printers' union, and the one to which the employers most strenuously object, is a demand for what is known as the "closed shop." The ultimate meaning of this expression is that the employing printer could not use in his composing room any man not a member of the union which makes the demand. It further requires that the union should have full authority in stipulating as to what individuals should be so employed. The whole thing has narrowed down into the question of whether a man shall be permitted to run his own business, or whether, on the other hand, he shall submit to having his business run according to the dictation of other parties interested in it only indirectly, taking a thoroughly selfish view of the situation and its requirements, and financially irresponsible, so far as the question is concerned of the placing of legal responsibility for acts committed. It seems to us, as to a great many others who have gone into this subject, that the whole idea of this suppression of individuality and substitution of human machines is entirely un-American in both conception and execution.

We do not wish to be understood as opposing the labor unions, for such is by no means our attitude. We believe that labor unions have done their members and the world in general a great deal of good, and that there is a large field for their continued efforts in this direction; but this fact does not blind our eyes to the equally well-known fact that in very many cases their demands have become not only exorbitant, but absolutely unjust and intolerable, and of such a nature as to give rise to serious doubts regarding their ultimate future. It goes without saying that if they force their employers into bankruptcy, or attempt to force them to accept conditions such as no self-respecting man can accept, their operations will act as a boomerang which will in time destroy themselves, and utterly nullify any usefulness which they may have had in the past.

The second paragraph of the Declaration of Independence states: "We hold these truths to be self-evident, that all men are created equal, that they are endowed by their Creator with certain inalienable rights; that among these are life, liberty and the pursuit of happiness."

It seems to us that the attitude assumed by the Typographical Union in this matter is distinctly at variance with the principle of granting to the employer either liberty or the pursuit of happiness. He is to be deprived of the liberty of conducting his own business in such a manner as to him seems right and just; he is to be deprived of the liberty of employing whom he sees fit; he is to be deprived of the liberty of disciplining employees whose work is below the standard in either quality or quantity. Being deprived of this liberty, it goes without saying that he is thereby deprived of the pursuit of happiness.

In referring to the quantity of work performed by an employee, it might be interesting to note that the rules which the union is attempting to foist upon the employers of this city prohibit the employer from molesting any member of the union (this would include any employee whatsoever, if the closed shop went into effect), for insufficient work in a given period of time, provided the extent of that work is equivalent to a figure stated by conservative authorities to be about fifty-five percent of that which could be done by a good compositor in the time in question. It is needless to state that such a provision as this conduces to laziness, breaks down the character of the man working under it, and might in easily conceivable cases operate to ruin the business of the employer.

The Merchant Marine.

Congress has under consideration the subject of the merchant marine. A bill has been introduced similar in all respects to the one which was side-tracked last spring, and based upon the report of the Merchant Marine Commission appointed more than a year ago. This bill looks to the up-lifting of the merchant marine by means of subsidies or subventions, call it what you will, given to vessels flying the American flag, and depending in amount upon the size of the vessel, and the length of time each year during which it is in actual operation. In addition to this, discriminating tonnage dues, favoring American shipping in the foreign trade, are suggested, the general trend of the total tonnage dues being such as to provide an increased revenue from this source, in order to defray to a certain extent the expense connected with the operation of the subsidy. Provision is also made for the enrollment of a force of naval volunteers, of constantly increasing proportions.

The feasibility of the subsidy is a question which has aroused a great deal of attention, and upon which honest differences of opinion are known to exist. The great difficulty in connection with the opposition to the subsidy lies in the fact that no substitute of any sort has been proposed in such a way as to demand much attention, with the sole exception of the discriminating duties advocated in the minority report presented to congress last spring. It is this very lack of alternative measures which has forced the issue between subsidy and no subsidy, rather than between subsidy and some other scheme for accomplishing the same purpose. There has been no serious disposition to belittle the requirements of the case, because those requirements are universally known and recognized. It has been simply a question as to the expediency of the methods proposed, and much of the question has related not so much to the mere fact of expediency as to the advisability of giving any sort of protection to a single industry.

It may be interesting to note that the virtual withdrawal of the United States from competition in the trans-oceanic commerce of the world has resulted in building up to an enormous degree the merchant marines of several of the other powers bordering on the Atlantic; and that several of these powers are at the present time adding very largely to the immense mercantile fleets which they already possess. During the present month a new ship-owning concern has been floated in Germany, with at first a moderate capital, but with ambitious plans for the future—plans which include a large extension of that capital as trade routes are opened up and developed, and the general trend of progress seems to warrant the increase. A statement issued by the Imperial Marine Office in Berlin, January 4, shows that the merchant marine of Germany advanced from 1,387,000 registered tons on January 1, 1896, to 2,533,000 tons at the present time. This represents an increase of 1,146,000 tons, or 82.6 percent; an increase greater by 250,000 tons than the entire merchant marine of the United States engaged in foreign commerce. The value of the German merchant marine ten years ago was $81,750,000. The present value is placed at $202,500,000. The gain is $120,750,000, or 147.6 percent. This gain was due entirely to the steam tonnage, which is, of course, the most valuable form, both from the point of view of economics and from that of national strength. The increase here was from $63,250,000 ten years ago to $188,250,000 at the present time, being an advance of $125,000,000 or 197.6 percent. During the same period, the value of sailing vessels decreased from $18,500,000 to $14,250,000, a loss of 23 percent.

Figures given by Lloyd's Register of Shipping show that on December 31, 1904, Germany had under construction 69 merchant ships of a total of 178,000 tons; The United States had under construction in coast yards 35 ships of 49,000 tons. Three months later, the German construction amounted to 87 ships of 205,000 tons; the American to 32 ships of 56,000 tons. At the end of June, the German construction was 77 ships of 227,000 tons; the American was 29 ships of 36,000 tons. September 30, 1905, Germany had under construction 72 ships of 225,000 tons; the United States, 26 ships of 38,000 tons. December 31, 1905, Germany had 94 ships of 270,000 tons; United States, 41 ships of 66,000 tons. In no case does the American tonnage above noted exceed 28 percent of that credited to Germany. On every one of the dates in question both France and Italy had a larger tonnage under construction than had the United States, the differences running from 13,000 tons March 31, in the case of each of the above powers, to 35,000 tons, in the case of Italy on June 30. On the latter date even Holland exceeded our construction by 8,000 tons.

We do not wish to institute any comparison between the tonnage under construction in the United States and that necessary to Great Britain. Aside from this one power, however, there is not a single nation in the world which has naturally so great an interest in sea-borne commerce as has the United States; not only are our exports and imports immeasurably superior in both bulk and value to those of any other power, but our coast line is enormous in extent, has many splendid harbors for the reception of ships, offers every facility which nature can offer to the proper development of a mercantile marine. It is simply a question of a difference in wages and in other operative costs between American ships and foreign ships; so long as this difference exists, it will be manifestly impossible to operate American ships in direct competition with those of other powers, and this condition will persist just so long as American standards of living remain superior to foreign standards of living, and no longer. In other words, the rehabilitation of the American merchant marine has come to be a question of either providing governmental assistance to shipbuilders and ship owners, or reducing the American scale of living to standards prevailing elsewhere in the world.

Which shall it be?

ABSTRACTS FROM RULES AND REGULATIONS OF THE GREAT LAKES REGISTER.

EDITION OF 1904.

ARRANGED BY HARRISON S. TAFT.

(*Continued.*)

2. Double reverse bars are to be fitted on their inner edge, extending from the center line to top of said web.

3. The web plates are to be worked intercostally between the side stringers, being double clipped to same with clips of reverse-bar size, and have in addition a diamond plate connection of the same thickness as the webs. When the depth of the web is 18 inches or below, the diamond plates are to be 24 by 20 inches, but 30 by 24 inches when said depth is above 18 inches.

4. If the web frames between decks are of increased depth they may be of reduced thickness.

44. Side Stringers.

1. Their depth and thickness are to be the same as that of the web frame plates.

2. When cut at a transverse bulkhead they are to be secured to same by a bracket plate of the same thickness as the stringer plates with double clips. Said brackets are to measure on each side at least three times the width of the stringer, being treble riveted to said stringer.

3. When worked continuously through a transverse bulkhead the width of the stringer plate at said point should be increased to double its ordinary width and have double staple work on each side of said bulkhead.

4. Said stringers are to be clipped to the shell plating between the frames; with at least seven rivets in each flange of the clip, when said clip is 4 by 4 inches or above; otherwise 6 rivets. The clips are to be worked double for three frame spaces on each side of a bulkhead.

5. There shall be an angle bar at the face of the reverse bars and on the inboard edge of the stringer and bracket plates unless said plates are flanged.

6. When said stringers are 18 inches or more in depth bracket plates are to be fitted to alternate frames so as to support said stringer plates.

7. See also Table 11.

45. Deck Beams and Knee Brackets.

1. Deck beams are to be of the same scantling as in ordinary vessels.

2. When cut in way of the expansion-trunk sides they are to be secured to same by bracket plates of the same size as those connecting the beams to the frames.

3. Strong beams are to be fitted to the "crown deck" to the head of all web frames, being secured to said web frames by large bracket plates.

4. Hatch-end beams are to be of increased size unless already augmented as stated in paragraph 3.

5. The knee brackets are to be at least three times the depth of the beams they are attached to in way of all oil compartments, having at least a six-rivet connection, spaced not over 4 1-2 diameters apart, to the frames.

6. See also Table 11.

46. Deck Stringers and Plating.

1. The deck stringers are to be worked continuously through all bulkheads. The intercostal clips connecting them to the shell plating are to have at least 6 rivets in each flange.

2. When the gunwale bar on the "crown deck" is worked continuously, it must be of increased size and double riveted in both flanges in way of all oil compartments.

3. In no case shall the "crown deck" be of less than 15 pounds plating, or the top of the expansion trunk of less than 12 1-2 pounds plating.

4. The deck plating must be increased in weight at all hatches, etc. When the upper-deck oil hatchways are worked continuously the strake of plating on each side of said hatchways should be worked double for 1-2 length amidships.

47. Stanchions.

Quarter stanchions are to be fitted throughout when the vessel's beam exceeds 42 feet. One stanchion only need be fitted in each tank unless the length of said tank exceeds 24 feet in a fore-and-aft direction, in which case there shall be two stanchions in each tank.

48. Shell Plating.

1. No reduction will be allowed in the after part of screw vessels up to the counter, when the machinery is located aft. Otherwise (except for the keel plates) the shell plating is to be as for ordinary vessels.

2. Overlapped butts in the way of the oil tanks should be at least treble riveted.

3. Butt straps in the way of the oil tanks are to be worked double and be treble riveted with three complete rows.

4. All longitudinal seams in way of the oil tanks are to be double riveted.

49. Engine and Boiler Space.

1. Web frames in said spaces are to be connected to the tank-top plating by double clips, with a gusset plate riveted to the reverse bars and tank-top plating.

2. When the machinery is located amidships a cofferdam is to be located at each end of said space.

50. Bulkheads.

1. Transverse Bulkheads.

(*a*) Said bulkheads should be fitted with single-frame angles, double riveted in both flanges.

(*b*) The bulkhead-shell liners need only be wide enough to take one row of rivets on each side outside of the frame angle bar.

(*c*) The horizontal stiffeners should be worked in way of the side stringers. They are to be built of plate with a face bar or a flanged edge, and be connected to the bulkhead plating by an angle bar. All as per Table 12. When said plates are 12 inches or more in width they are to be supported by brackets of the same weight as the stringer plates, spaced not more than 6 feet apart.

(*d*) The vertical stiffeners, worked on the opposite side from the horizontal stiffeners, should have bracket plates at their upper and lower ends, connecting said stiffeners to the deck and shell plating respectively.

(*e*) When the vessel's beam exceeds 36 feet a web plate of the same weight as the bulkhead plating is to be fitted to the vertical stiffener, midway between the center line and the sides of the ship on each side. Said web plate is to be at least 24 inches wide at the lower edge, with double angle bars fitted to its outer edge. Said web is to be secured to the deck plating and floors by bracket plates.

(*f*) When the breadth exceeds 42 feet two such webs are to be fitted on each side of the center line.

(*g*) The cofferdam bulkheads must be securely connected to each other by bracket plates, etc. Large bracket plates are to be fitted on the outside of said bulkheads at the floor and bilge stringers.

(*h*) No opening will be allowed in the cofferdam bulkheads below the "crown deck." They are to be treated as an oiltight compartment in all cases.

2. Center-Line Bulkheads.

(*a*) The weight of plating and stiffener scantlings are to be as given for the athwartship bulkheads in Table 12, except for the bottom strake, for which see section 38, paragraph 2.

(*b*) The vertical stiffeners are to be spaced as per frames, being connected at their lower ends to the floor brackets at the center-line bulkhead, and having similar brackets connecting their upper ends to the deck beams.

(*c*) Horizontal-plate stiffeners should be worked in way of the athwartship-bulkhead horizontal stiffeners, being connected to said horizontal stiffeners by bracket plates of the same weight as the plate stiffeners, with a breadth each way measured from the bulkhead of three times the width of the plate stiffener. A bracket plate should be worked on the opposite side of the bulkhead in way of aforesaid horizontal stiffeners.

(*d*) All plating above the lower strake may be worked intercostally between the athwartship bulkheads or continuously through them as desired.

(*e*) A vertical plate stiffener, equal to the weight of the bulkhead plating, is to be fitted the full depth of said bulkhead in way of the web frames at the side of the vessel. Said web plate shall be at least 2 feet wide at the lower edge, with a face angle of reverse-bar size, and be bracketed to the floors and deck beams.

(*f*) The extreme ends of a longitudinal bulkhead must taper for a distance at least equal to 1-3 of its depth from the keel to the "crown deck."

(*g*) When said bulkhead extends up into the expansion trunk, it may be 2 1-2 pounds less in weight within said space than the trunk's side plating, with stiffeners of reverse-bar size spaced as per frames, unless the bulkhead is made oiltight, in which case its scantling and that of its stiffeners must equal that of the trunk side plating.

3. It is recommended that the bulkhead plating be caulked on both sides, together with their stiffener angles.

4. Oiltight bulkheads must be made oiltight without the use of felt, packing, or lead.

TABLE 12.—SCANTLINGS FOR LONGITUDINAL AND TRANSVERSE OIL-TIGHT BULKHEADS.

Numbers B + D	Weight of Plating. Lbs.			Horizontal Stiffeners.		Vertical Stiffeners.		
	Lower Third	Middle Third.	Upper Third.	Breadth and Weight of Plates.	Size of Angles.	Bulb Angles. Size.		Maximum Spacing.
42 to 46	15	14	12½	9 inches × 14 lbs.	3 × 3 inches × 7.2 lbs	5 × 3 inches × 9.8 lbs.		24″
46 " 52	16	15	14	10 " × 15 "	3 × 3 " × 7 2 "	5½ × 3 " × 12.3 "		24
52 " 56	17	16	15	10 " × 16 "	3½ × 3½ " × 8.5 "	6 × 3 " × 12.3 "		25
56 " 60	18½	17	15	11 " × 17 "	3½ × 3½ " × 8.5 "	6 × 3 " × 13.75 "		25
60 " 64	20	18½	16	12 " × 18½ "	3⅜ × 3½ " × 8.5 "	6½ × 3½ " × 17.2 "		26
64 " 67	20	18½	16	13 " × 18½ "	3⅜ × 3½ " × 9.8 "	7 × 3½ " × 18.0 "		26
67 " 70	21½	20	17½	14 " × 20 "	3½ × 3½ " × 9.8 "	7 × 3½ " × 19.25 "		27
70 and above	21½	20	17½	15 " × 20 "	3⅜ × 3½ " × 9.8 "	7 × 3½ " × 19.25 "		27

51. Expansion Trunks.

1. The plating of said trunks is to equal that of the upper part of the tank bulkhead, as per Table 12.

2. The stiffeners are to be of the same scantling as the frames and be located in way of the deck beams. They are to be bracketed to said beams at both top and bottom by knees of the same thickness as the flange of the stiffeners to which they are attached.

3. Web plates of the same thickness as the trunk side plates are to be fitted at the ends of all hatchways.

4. Fore-and-afters are to be fitted in way of all half-beams where the deck plating is flanged in way of an expansion trunk.

52. Riveting.

1. Rivets of less than 3-4-inch diameter will not be allowed in oil tanks subject to pressure.

2. Widths of butt straps, etc., to be the same as for ordinary vessels, as per Table 7 and sections 20 and 21 of Part I., except as noted below.

3. Single-riveted longitudinal shell seams will not be allowed in way of an oil compartment.

4. If the center-line bulkhead is not made oiltight, all seams and butts of same, except those of the bottom strake, can be single riveted.

5. Butts of the "crown-deck" plating are to be double or treble riveted as called for.

6. The Following to be Double Riveted:

(*a*) Butts and seams of all oiltight bulkheads. Chain fashion.

(*b*) Upper edge of the lower strake of plating of the center-line bulkhead in all cases.

(*c*) Oiltight-bulkhead frame bars; corner angles; and collars subject to oil pressure. Zigzag fashion.

(*d*) Longitudinal seams of the "crown-deck" plating. Chain fashion.

(*e*) Knee brackets; bracket plates in general when worked to maintain continuity of strength. Zigzag fashion.

(*f*) Single frame bars at web frames. Zigzag fashion.

(*g*) Web-frame butts.

7. The Following to be Treble Riveted:

(*a*) All butts of the lower strake of a center-line bulkhead.

(*b*) Web frames to floor plates.

8. In oiltight bulkheads and in all other parts, where the oil pressure comes from both sides, and in cases where more than two thicknesses of plates are riveted together, it is recommended that rivets with countersunk heads as well as points be used, or else the plug-neck type of rivet be substituted. The head of the rivet should project beyond the plate at least 1-4 of an inch, with sufficient spread on said head to prevent it from being drawn.

53. Spacing of Rivets.

	DIAS.
1. Longitudinal seams of the shell plating; butts and seams of the "crown-deck" plating; butts and seams of all bulkheads made oiltight; chain riveted	3½
2. Oiltight-bulkhead frame bars; corner angles and collars; bulkhead stiffeners to be caulked	4
3. Butts and seams of a center-line bulkhead not made oiltight; deck-beam knee brackets; all bracket plates worked to maintain continuity of strength	4½
4. Bulkhead stiffeners not caulked	6
5. Frame and deck-beam rivets	6½
6. Single frame bars at web frames. Zigzag riveted	7

7. Side-stringer shell clips; 7 rivets in each flange when of 4 by 4 inches, but 6 rivets if of less size.

8. Deck-stringer shell clips; 6 rivets in each flange.

9. Deck-beam knees; 6 rivets in each knee.

See Table 7, Part I.

PART III.

54. 36-Inch Frame Spacing.

1. The extent of a 36-inch frame spacing is to be from the forward collision bulkhead to the bulkhead at the forward end of the engine room when the machinery is located aft. When the machinery is located amidships said 36-inch frame spacing can be worked throughout said machinery space.

2. When the engines are located aft the frame spacing throughout the engine room and aft peaks must not exceed 24 inches.

3. The regular frame spacing in the fore peak must not exceed 24 inches with intermediate frames as in ordinary vessels (Section 6, Part I.).

If the regular frames are spaced 18 inches apart these intermediate frames may be dispensed with.

4. With a 36-inch frame spacing the shell plating must not be less than 3-8 inches thick (15 pounds). When the shell plating is less than said thickness the frames must not be spaced more than 24 inches apart.

5. The frames, floors, and reverse bars throughout the engine-room space, when located aft, are to be as in ordinary vessels.

6. (*a*) When the sum of the beam and depth (B + D) is under 82, web frames will not be required outside of the engine-room space; but it is recommended that web frames, spaced not over 15 feet apart, be fitted in the forward cargo hold when said sum is 76, 78, or 80.

(*b*) When said sum is 82 and 84, the above web frames are to be fitted throughout the cargo space, being spaced 24 feet apart. When said sum is 86 and above, they are to be located at each hatch beam.

(*c*) The web frames in the engine room are to be spaced not over 12 feet apart.

7. The spacing of the hold stringers, and the between-deck side stringers, should be such that they will not be more than 4 feet 6 inches apart or from a deck amidships; so that the area of unsupported shell plating at the forward and after collision bulkheads will not be over 18 superficial square feet, said stringers being chocked to the shell. When said area is over 20 superficial square feet, panting stringers must be fitted.

PART IV.

55. Stowage of Grain.

1. When the vessel is not fitted with a double bottom, a grain-tight raised platform must be fitted a sufficient distance above the floors, so as to allow a free flow of the bilge water to the pumps. In flat-bottom vessels said platform must be 10 inches above the floors at the center line and 12 inches at the bilge.

2. Vessels of 35- to 40-foot beam are to have a center-line shifting board extending from the under side of the upper-deck beams to half depth of the hold. When built of wood they are to be at least 1 inch thick, being secured to the center-line stanchions. When said stanchions are spaced over 4 feet apart the shifting boards must be made thicker in proportion.

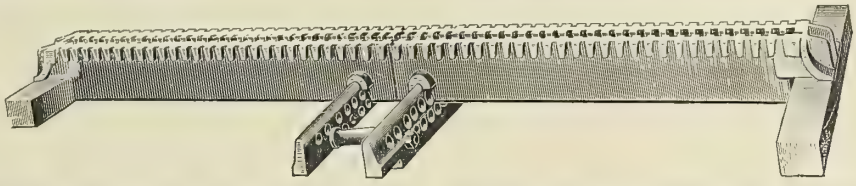
3. When said beam is 40 feet and above, two such longitudinal lines of shifting boards are to be worked, each located midway between the center line and the side of the ship, the above center-line shifting board being omitted.

4. When a second deck is fully laid and depth of hold below said deck exceeds 10 feet, a center-line bulkhead or partition should be fitted below said deck to 1-2 depth of hold.

ENGINEERING SPECIALTIES.

The Barker Grate Bar.

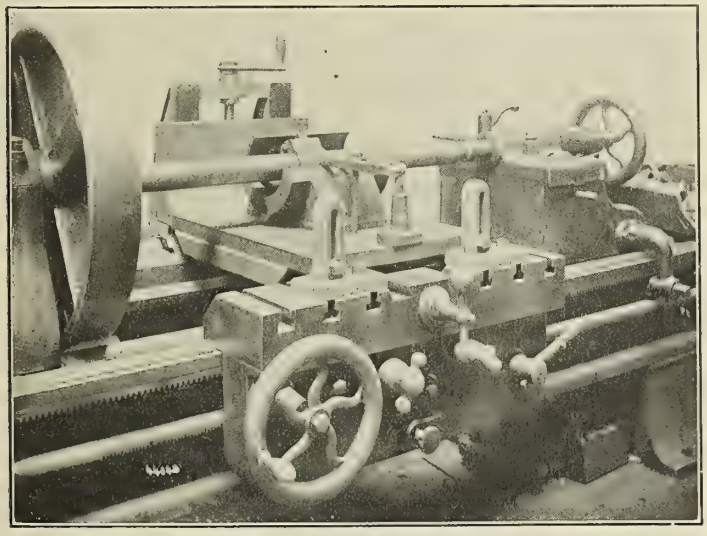
In this bar the first advantage to be noted is the great percentage of air space and its even distribution over the entire grate surface. The air is not obstructed by any braces, spacing lugs or blank spaces on the grate surface, there being a clear, open line of air between the bars from one end to the other. The illustration shows clearly these important features. The bar is one which offers much less obstruction to combustion than other kinds and as a result effects a saving of coal that soon makes the first cost a secondary consideration.



It is a twin bar with the lower edge of the webs united by two braces, placed well below the heated surface and consequently always cool. The braces hold the bars rigidly against warping, and yet do not interfere with the free passage of air. In order to avoid the possibility of the rake catching in the joint between the front and rear sections of the bars where they meet over the middle bearing bar, one bar in each twin bar is made slightly longer than the other, so that when the complete grate is assembled a zigzag joint is formed across the grate. The bar is placed upon the market by the Robsmith Specialty Company, Provident Building, Philadelphia.

Lathe Attachment for Boring.

An attachment for converting a lathe into a horizontal boring machine is the new product of H. B. Underwood & Co., 1025 Hamilton street, Philadelphia. The attachment is placed in the position ordinarily occupied by the cross slide, which must be removed when the lathe is to be used as a boring machine. It serves the same function as the work table of a horizontal boring



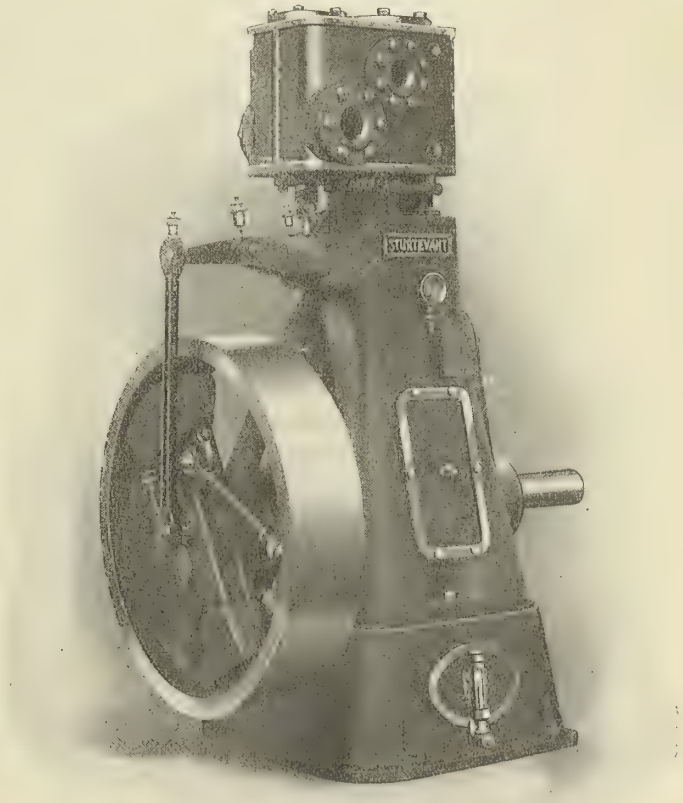
machine and is adjustable laterally and vertically. The boring tool is mounted on a bar supported between the head and tail of the lathe. It is claimed that the work done is in every respect equal in quality and quantity to the output of a regular boring mill of equivalent capacity.

The attachment consists in the main of two castings; one, resembling in miniature the housings of a boring mill or planer, is attached to the rear of the lathe carriage, and the other is a horizontal table plate having a vertical leg adapted to slide on ways planed on the rear casting. The surfaces of this angle plate are planed square or parallel with all the other parts of the attachment. The vertical adjustment of the table plate on the supporting casting is effected by an adjusting screw. The bed of an attachment for a 30-inch lathe has three T slots for ⅝ or ¾ inch bolts, is 31 inches long by 22½ inches wide and has a vertical adjustment of 5½ inches.

This attachment may be readily fitted to any lathe, it being necessary only to remove the tool slide and drill and tap the holes in the back of the carriage for securing the upright casting. Thereafter the time required to change the lathe from a turning to a boring machine is covered by a very few minutes. The weight of the attachment for a 30-inch lathe is 685 pounds. For this size a 3 or 3¼ inch boring bar with suitable cutter heads and automatic feed is recommended.

Sturtevant Vertical Engines.

One of the features of the Sturtevant vertical engine (Hyde Park, Mass.) is a system of forced lubrication, which, with the



complete inclosure of the moving parts, provides for continuous operation for weeks at a time without attention, and it is said insures

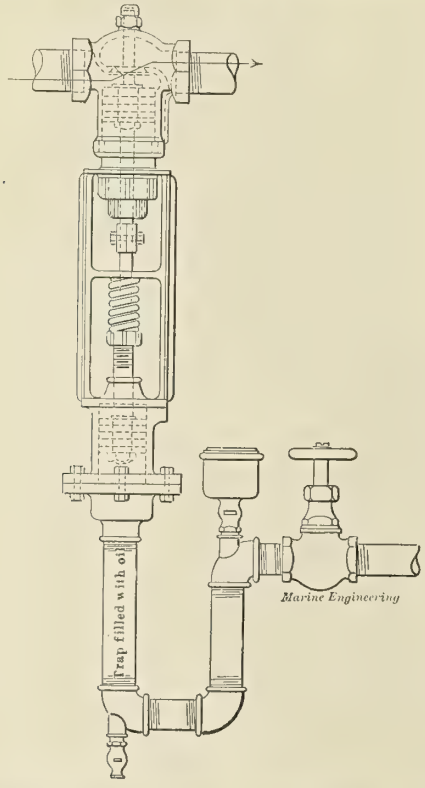
perfect reliability even in the hands of the unskilled. This type of engine is adapted to all classes of work requiring maximum power in minimum space, a condition attained by operation at the high speed made possible by refined design, high-grade construction and the use of the best material. They are especially designed for the driving of direct-connected generators. Because of the large diameter and short stroke, they develop great power and high rotative speed without excessive piston travel.

Within the heavy cast-iron base, to which is bolted the lower part of the frame, a submerged oil pump, operated by the crank shaft, draws oil from the reservoir and forces it through pipes and internal passages in the moving parts to the crank pin, the wrist pin, and the main bearings. Twice during each revolution the reversal of stress on these parts, due to the double-acting feature of the engine, so reduces the pressure that the pump has an excellent opportunity to force between the surfaces a fresh film of oil, which is carried around to lubricate the rotating parts, when the pressure is greater. The pressure of 10 to 20 pounds per square inch maintains this film of oil, preventing actual contact with metal, reducing wear and friction to a minimum, and insuring a mechanical efficiency of over 90 percent.

The engine is built in sizes ranging from 6 to 85 horsepower, depending upon the steam pressure employed. The weights vary from 1,000 to 4,300 pounds, and speeds of revolution from 475 per minute in the smaller sizes to 275 in the larger.

The Ideal Automatic Pump Governor.

For the purpose of maintaining a pump in a condition always ready for any emergency this governor has been placed upon the market by the Ideal Pump Governor Company of 15 Whitehall street, New York. One of the main features consists in the fact that the piston and water pressure cylinder are protected from corrosion and from sticking by means of a body of oil which follows the piston on both strokes, and thoroughly lubricates all

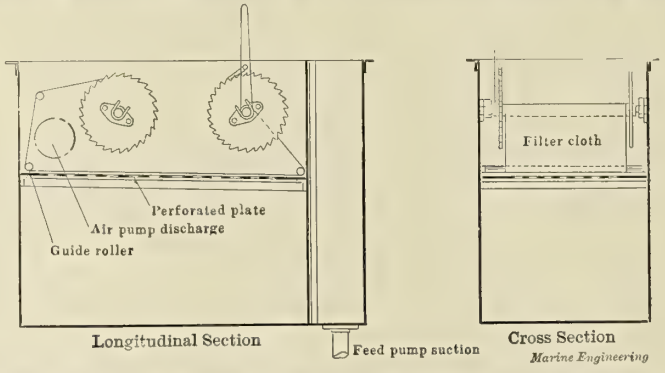


wearing surfaces in the cylinder. This makes it virtually impossible for the pump to stick at a time when an emergency requires that it be used, as well as provides a great saving in wear and tear and in consequent repairs. The spring shown in the illustration operates to start the pump working at any time when the pressure in the stand or delivery pipe falls below a predetermined value, and when this pressure reaches the proper figure the oil is pressed against the piston and the throttle valve shut against the pressure spring. It is seen that this action is automatic.

An illustration of the value of this sort of equipment was afforded early in December when the steamer *Rosalie* of the Joy Line was discovered to be on fire in the berth next to that occupied by the United States transport *McClellan*. On the latter ship this governor was fitted, and operated to govern the pump in such a way that it was not necessary to pass word back and forth between those fighting the fire and those in the engine room of the transport, it being found that the pressure, which was set for 100 pounds per square inch, was easily maintained on the line until the fire was out. No time was lost in getting the pumps started, it being necessary simply to connect the hose and open the stop valves.

The Milne Filter.

By means of a roll of towelling, or any other fabric which will allow water to pass freely through it, a simple filter arrangement has been perfected and patented by J. S. Milne of the Neafie and Levy Shipbuilding Company of Philadelphia, and tested upon the steam yacht *Josephine*. The filter cloth is stretched across a wire screen or perforated plate, with plenty of lap at the ends and sides, and, being provided in a roll of thirty or forty yards in length, it is seen that once installed the equipment is virtually permanent for a considerable length of time.



When the cloth becomes so saturated with material taken out of the water that the latter will not readily pass through it, a handle is turned, shifting the saturated portion to a receiving drum and supplying new material for the operating portion of the filter. It is estimated that under ordinary conditions a roll will last from forty to fifty days of continuous steaming, with little more attention than the changing of the cartridge. Tests have shown that this filter catches practically every bit of the oil and other impurities in the water, as well as matter in suspension, such as mud and sand. It can of course readily be made automatic by attaching to the receiving roll either clock work, or some other motive device.

TECHNICAL PUBLICATIONS.

The Shipbuilding Industry; Its History, Practice, Science and Finance. By David Pollock, M. I. N. A. Size 5 by 7½ inches. Pages 199. 1905, Methuen & Company, London. Price 2/6 net.

Beginning with the earliest history of shipbuilding, dated far back of the time of Noah's Ark, and carrying the reader through the Roman and Norseman epochs, this work covers the history of the subject from the very beginning up to the present time, giving quite a little information regarding early questions relating to steam propulsion, and the first successful examples of steam vessels.

The second chapter is devoted to the development of the modern steamship, from about the period of the *Great Eastern*, giving a number of illustrations, nearly all half tones, showing various features of the ships and of their construction, and carrying the subject up to the advent of the marine steam turbine in large ships. The next subdivision covers the subject of shipbuilding in up-to-date ship yards, going into some details regarding the preparation of the frames and plates, together with the subjects of punching, shearing, riveting and the crane equipment of the yards. This gives also some information regarding model tanks and their uses. The fourth chapter is devoted to evolution in structures and types of ships, and includes comments upon the types known as turret ships, together with the use of steam turbines as a means of propulsion. The last chapter deals with the volume and sources of output and the general classes of shipping forming that output, covering all the tonnage-producing districts and countries on the globe. In this chapter are given tables of production covering a number of years, not only in Great Britain, but in all the other countries in which shipbuilding is carried on. The illustrations are all printed on inset sheets, and with two exceptions consist entirely of half tones. They are very well chosen and contribute much to the value of the work. There is one folder giving a longitudinal section of the new Cunard ship *Caronia*. The good index at the rear of the volume ought to be appreciated.

Elementary Mechanics. By George A. Merrill, B. S. Size 5½ by 8 inches. Pages 267. Illustrations 175. 1905, New York. American Book Company. Price $1.50.

Intended for the upper classes in elementary schools and the lower classes in college, this book is essentially an elementary text book, presupposing the completion of work in trigonometry, but not requiring for its development the use of the calculus. It has, of course, nothing like the scope of the larger and more pretentious works on the subject such as are used in our large engineering colleges, but it is much more adequately prepared for the purpose than is the case with the portion of the physics works devoted to the subject of mechanics, even including those used in colleges. It is subdivided into the logical sections of kinematics, statics and kinetics; under each of which heads the explanations and proofs are given with a great deal of care, and examples are given covering the range of the material in the text. The illustrations, almost entirely linear, are clear and well chosen for the purpose.

Record of American and Foreign Shipping. Pages 1,322. Size 8 by 9½ inches. Price $15.00. American Bureau of Shipping. 66 Beaver street, New York.

The volume for 1906 of the "Record of American and Foreign Shipping" (American Lloyd's), which is the thirty-eighth annual issue of this valuable register, is now being delivered to subscribers.

The Record contains full reports and particulars of 14,645 vessels, ranging from the ketch to the full powered trans-Atlantic liner, and flying the flag of every nation, alphabetically arranged, with much detail as to build, ownership, and condition. This information forms the bulk of the annual volume, but it also contains rules for the construction and classification of all classes of vessels, with illustrations and tables of great technical and practical value; revised rules for the construction of machinery and boilers, electric installations and refrigerating apparatus on shipboard.

The volume contains names of vessels which have been changed; list of compound names arranged alphabetically by last word of name, for ready reference; list of addresses of prominent shipbuilders, drydocks and marine railways of the United States; list of owners of vessels, all of much value to the shipping interests. The work is approved and indorsed by the important boards of underwriters of the United States, and is accepted by the merchants and underwriters throughout the world as a standard classification of shipping. Supplements to the volume, issued semi-monthly, keep subscribers informed of new vessels constructed during the year, with reports of repairs to old vessels, and other useful information.

QUERIES AND ANSWERS.

All reasonable questions concerning marine engineering received from and signed by subscribers will be answered by the Editor in this column. All communications must bear the name and address of the writer.

Q. 310.—Please answer the following questions regarding good practice in the design for purpose of cooling any space:

(1) How much pressure is necessary to liquefy carbonic acid gas at 65 degrees Fahrenheit.

(2) If copper and iron pipe be acted upon by carbonic acid gas, by what means can it be protected?

(3) Under what pressure may the carbonic acid gas on the market be stored?

(4) How much quantity of gas can be accumulated to a jar?

S. W.

A.—(1) The pressure required to liquefy carbonic acid gas ranges from about 680 pounds per square inch at 50 degrees Fah. to 1,000 pounds at about 80 degrees Fah. At 65 degrees the pressure required will be about 830 pounds per square inch.

(2) Carbonic acid does not attack copper or iron, and this fact, together with its lack of chemical activity toward most food substances is one of the points of advantage which this substance may claim as a refrigerating agent.

(3) The answer to (1) will sufficiently answer this question.

(4) The jars or cylinders usually contain from 20 to 22 pounds of liquid carbonic acid.

Q. 319.— Will you place me on the right track as to the calculations for a gasoline engine and let me know how the following elements are obtained? The engine is to be of the four-cycle type with three cylinders, 5-inch bore, 5-inch stroke; 800 revolutions per minute.

(1) Area, or depth of the compression space.
(2) Area in square inches of the admission valves.
(3) Area in square inches of the exhaust valves.
(4) Thickness of cylinder walls.
(5) Length of connecting rod.
(6) Approximate horsepower.

A.—(1) The volume of the compression space will be from 30 to 35 percent of the piston displacement; in this case being about 33 cubic inches for each cylinder.

(2) The area in square inches of the admission passage will be represented by the formula $a = \frac{NV}{36{,}000}$, where N represents the number of revolutions per minute and V is the piston displacement in cubic inches. In the example in question $a = 2.2$ square inches, representing a diameter of nearly 1¾ inches.

(3) The area of the exhaust passage should be 30 or 40 per cent greater than that for admission, though in many cases they are made of the same size. For the present case this would figure out at 3 square inches, giving a diameter of 2 inches.

These three questions were answered at some length on page 441 of MARINE ENGINEERING for October, 1905.

(4) An expression for the thickness of cylinder wall is given on page 153 of Lucke: "Gas Engine Design," as $t = 0.075d$. In the present case this figures out at ⅜-inch.

(5) The length of connecting rod is commonly made from five to six times the length of the stroke. In the present case that would mean from 13 to 15 inches.

(6) The approximate horsepower is found from the formula $H = \frac{PLAN}{33{,}000}$. In the present case the result is about 24.

Q. 324.—Will you please give me a formula for determining the correct equivalent mean effective pressure in the low-pressure cylinder of a compound or triple expansion engine? M. E. P. $= \frac{p\ (1 + \text{hyp. } r)}{r}$ gives the theoretical value, while some authorities give $\frac{.9 \times p \times (1 + \text{hyp. } r)}{r} - .9p$

as the real value, but this I find about 20 to 30 percent too high, according to results given in tables. J. G. R.

A.—For a single cylinder engine the M. E. P. is found from the expression $p \frac{(1 + \text{hyp. } r)}{r} - p$ where p is the initial pressure of the steam in pounds per square inch, p_b is the back pressure of the steam, and r is the ratio of expansion in the cylinder.

For an engine having more than one cylinder, the mean effective pressure as reduced to the low-pressure cylinder is found from the same expression, but in this case it is necessary to employ a reduction factor to cover the fact that from one cylinder to another there is a loss in effective pressure due to the necessary pressure gradient for carrying the steam through the steam passages, and a further loss due to condensation of the steam in those passages and to radiation. General practice places the multiplying factor for a compound engine at from 0.8 to 0.9, with an average of perhaps 0.85; for a triple expansion engine this "card factor," as it is called, becomes from .65 to .75, with an average of perhaps .7; the corresponding figure for a quadruple expansion engine is placed at .55 or .60; the exact value in each case being determined by a consideration of the qualities of the engine with regard to design.

Q. 327.— There is a watertight compartment 12 by 12 feet and 8 feet high through which the rudder post, 21 inches in diameter, passes. The stuffing box, which is 7 feet below the surface of the water, is leaking so that it takes a pipe 1½ inches in diameter to drain it off. As they do not want to dock the ship the engineer in charge proposes to pump air into it at such a pressure as will overcome the leakage by forcing the water back through the stuffing box, while a man stays in there and packs the gland. Would you kindly answer, what pressure would be required to do it; and against what pressure of air a man could work while in a place like that? This occurred on board a warship.

W. H.

A.—If we understand the question aright, the head of water under which the leakage occurs is 7 feet, and the pressure due to such a head of water figures out at a trifle more than 3 pounds per square inch. It is quite a common practice for men in constructing submarine tunnels, such as those now being driven beneath the Hudson river and East river at New York, to work in a pressure amounting to one atmosphere above the regular air pressure, and in many cases it is necessary to increase the pressure in the compartments in which the men are working to two or more atmospheres. One atmosphere represents a pressure of 14.7 pounds per square inch, which is nearly five times that required in the case mentioned, and while the effects are not particularly bad, yet there is a certain amount of discomfort connected with such a pressure. This would probably not be the case with any pressure less than 6 or 8 pounds per square inch. The scheme proposed for the stoppage of the leakage is not only feasible, but is one which has been used for other purposes in a great many cases.

Q. 328.— There is a gas engine with cylinder diameter 2⅝ inches and a 3-inch stroke. The combustion chamber contains 2.55 cubic inches. There are two balance disks weighing 4 pounds each and being 6 inch in diameter. What is the proper size and weight for a fly wheel? The engine is supposed to run 300 to 400 revolutions.

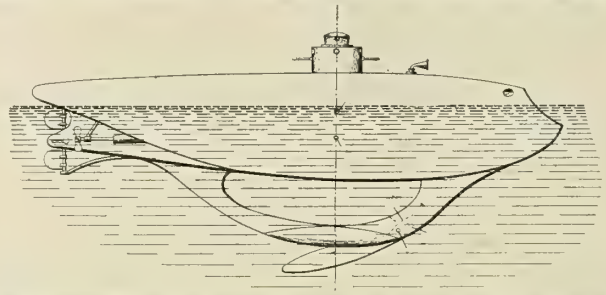
P. A. DeL.

A.—Without knowing something about the action of this engine with regard to smoothness of working and particularly with regard to the crank effort diagram, it would be out of the question to give an answer to this query. Makers of engines of this character doubtless fit fly wheels which have been found by practice to cover the case sufficiently well, without taking the trouble to go into the mathematics of the question and determine accurately the proper size and weight for such a wheel, and such a method of procedure gives quite satisfactory results, although in many cases it is a question of guesswork rather than a real knowledge of the requirements of the case. The information given by our correspondent is not sufficient for a mathematical determination of the size and weight of fly wheel, because, as stated above, it would be necessary to know the exact history of the crank effort through the continuance of a complete cycle; and to know further within what limits of variation in momentary speed it was proposed to run the engine.

SELECTED MARINE PATENTS.

803,172. SUBMARINE VESSEL. SIMON LAKE, BRIDGEPORT, CONN.

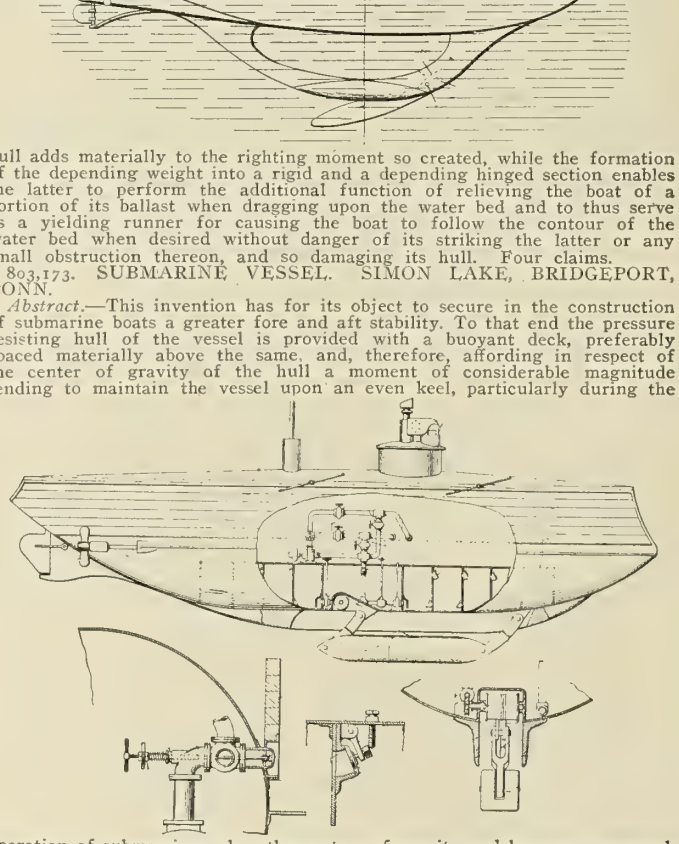
Abstract.—The invention consists, essentially, in the combination, with the hull, of a depending web or support which sustains a weight at some distance below the lower side of the hull and beneath its center of buoyancy. By locating the weight outside of the hull and maintaining it in a fixed relation thereto the center of gravity of the boat as a whole is lowered to such a degree beneath the center of buoyancy that a decided tendency is created to preserve the boat in its normal fore and aft trim and which opposes any relative vertical movement by the bow or stern during submergence. The addition of a conning tower upon the upper side of the



hull adds materially to the righting moment so created, while the formation of the depending weight into a rigid and a depending hinged section enables the latter to perform the additional function of relieving the boat of a portion of its ballast when dragging upon the water bed and to thus serve as a yielding runner for causing the boat to follow the contour of the water bed when desired without danger of its striking the latter or any small obstruction thereon, and so damaging its hull. Four claims.

803,173. SUBMARINE VESSEL. SIMON LAKE, BRIDGEPORT, CONN.

Abstract.—This invention has for its object to secure in the construction of submarine boats a greater fore and aft stability. To that end the pressure resisting hull of the vessel is provided with a buoyant deck, preferably spaced materially above the same, and, therefore, affording in respect of the center of gravity of the hull a moment of considerable magnitude tending to maintain the vessel upon an even keel, particularly during the



operation of submersion, when the centers of gravity and buoyancy approach each other somewhat under the changing conditions occurring in the course of such operation. To insure a comparatively large displacement for the upper portion of the vessel, not only is the deck preferably constructed of wood planking suitably calked to insure against penetration of water, but the sides of the superstructure are made of like material, thus providing an entire superstructure of buoyant material and preferably of buoyant construction. Fourteen claims.

803,174. BALLAST DEVICE FOR SUBMARINE VESSELS. SIMON LAKE, BRIDGEPORT, CONN.

Abstract.—According to the present invention the vessel is provided with a flat guide plane or planes substantially parallel with the direction of thrust of its propelling means and comprising a comparatively flat deck or lateral guards or fins along the edges, or both, in combination with a heavy ballast weight which is movable longitudinally in respect of the center of gravity of the hull, the shifting of which forwardly or backwardly operates to give the vessel a slight upward or downward inclination in opposition to the steadying tendency of the guide plane or planes, and thereby cause the vessel to rise toward or descend from the surface. By suitably suspending the ballast weight it is also adapted to serve the additional function when lowered materially beneath the hull into its normal position beneath the center of gravity of correspondingly lowering the resultant center of gravity of the component parts of the structure. Six claims.

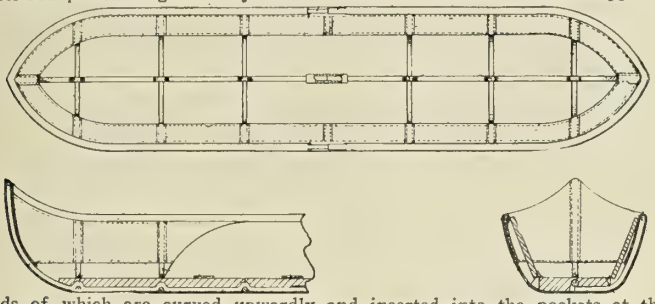
803,175. BALLAST APPARATUS FOR SUBMARINE VESSELS. SIMON LAKE, BRIDGEPORT, CONN.

Abstract.—The present invention further provides for the positive cutting off of communication between the filled and unfilled or partially filled tanks of the system by locking the intervening overflow check valve, so that the pitching or careening of the vessel from any cause shall not effect the shifting of the water ballast to any appreciable extent, and thereby injuriously affect the equilibrium of the vessel. Thirteen claims.

803,272. BOAT. JOHN F. CONKLING, HAWLEY, PA.

Claim.—3. A boat, the body portion of which is composed of flexible material and provided at the top of the side and end portions thereof with pockets, the opposite sides of the boat being also provided with transversely arranged pockets, top side rods composed of separate parts inserted into the

top side pockets centrally of the boat and the opposite ends of which are curved inwardly, and the adjacent ends thereof rigidly connected, a bottom keel rod placed longitudinally of the bottom of the boat and the opposite

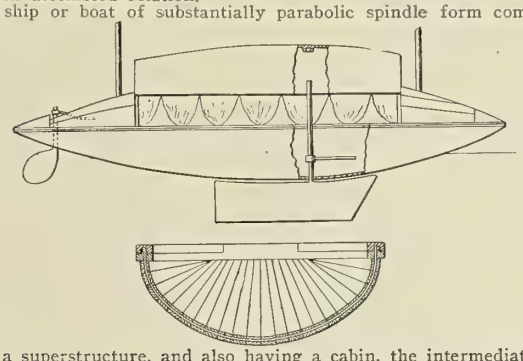


ends of which are curved upwardly and inserted into the pockets at the opposite ends of the boat, said bottom keel rod being also composed of separate parts rigidly connected and transverse ribs or knees the ends of which are curved upwardly and inserted into transverse pockets in the opposite sides of the boat, said boat being also provided with a supplemental bottom which rests on the bottom keel rod and transverse ribs or knees, and which is composed of two parts hinged together. Four claims.

803,607. BOAT OR SHIP. CHARLES P. LEAVITT, MIAMI, FLA., ASSIGNOR OF TWO-THIRDS TO ARVID P. CURRY, MIAMI, FLA.

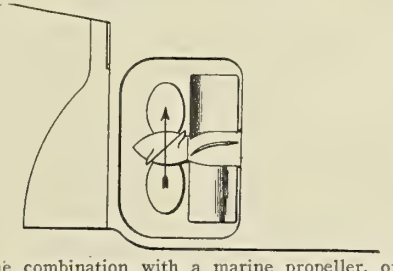
Claim.—1. A ship or boat of substantially parabolic spindle form, comprising a hull and a superstructure, each composed of a plurality of staves arranged side by side, and rods extending transversely of the hull and superstructure and constituting the sole means for holding the staves together in assembled relation.

5. A ship or boat of substantially parabolic spindle form comprising a



hull and a superstructure, and also having a cabin, the intermediate part of the superstructure constituting a roof for the cabin, divided bulkheads in the ship or boat, one section of each being connected with the intermediate part of the superstructure and the other part with the hull, and means for raising said intermediate part of the superstructure. Nine claims.

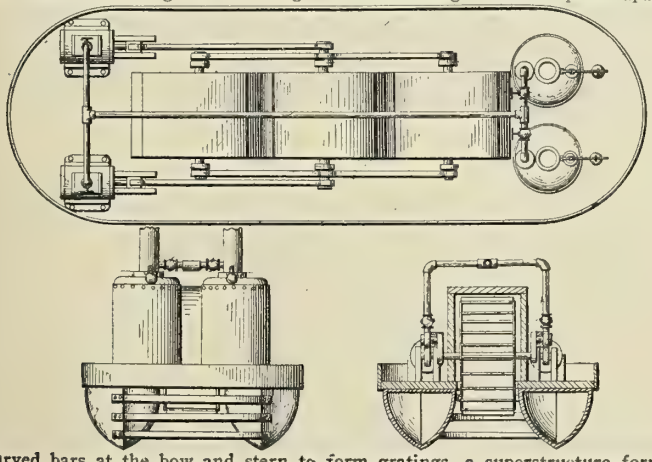
803,671. MARINE PROPELLER. CHARLES G. CURTIS, NEW YORK.



Claim.—1. The combination with a marine propeller, of curved or inclined stationary guide-blades located in front of the propeller and radiating from the propeller-shaft. Five claims.

803,768. STEAMBOAT. JEREMIAH LOCKWOOD, SULLIVAN, IND.

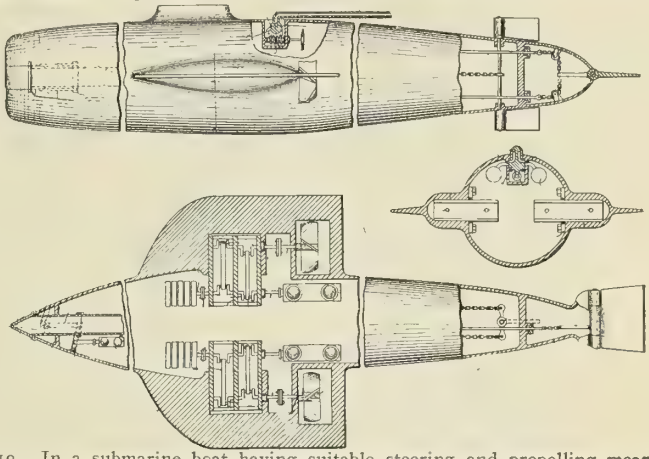
Claim.—In a boat of the character described, the combination with two hulls, each formed independent of the other and of air-tight construction, of a deck connecting said hulls together and holding the same spaced apart,



curved bars at the bow and stern to form gratings, a superstructure forming a paddle-wheel box on said deck, independently-mounted paddle-wheels arranged in said paddle-box to turn between said hulls, a removable cover arranged over each of said wheels, cranks arranged on the ends of said paddle-wheel shafts, connecting-rods connecting said cranks together, engines arranged on said deck, links to connect the piston-rods of said engines to said connecting-rods whereby said paddle-wheels are simultaneously driven, and boilers arranged on said deck to supply steam to said engines. One claim.

803,885. SUBMARINE BOAT. HALVOR O. EIANE, OF THE UNITED STATES NAVY.

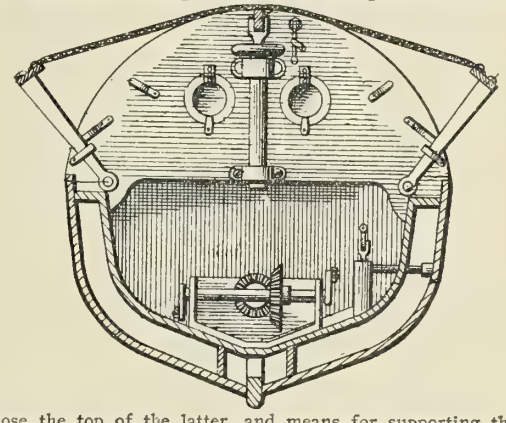
Claim.—1. In a submarine boat or vessel, the combination of a hull, laterally-disposed hydroplanes beneath the surface of the water, propellers arranged at the sides of the vessel, said hydroplanes being chambered and removable crank cases in the chambers of said hydroplanes and accessible from inside the hull, connections between said crank cases, the propellers and a source of power.



10. In a submarine boat having suitable steering and propelling means, a torpedo tube opening at the prow of the vessel, a closure for said tube, said closure comprising two hinged jaws co-operating to fold over the end of the tube, said jaws having slotted crossed rearward extensions, and means for operating said extensions from the interior of the vessel to open and close the jaws simultaneously. Ten claims.

804,095. LIFE-BOAT. FREEMAN W. BROWN, TACOMA, WASH.

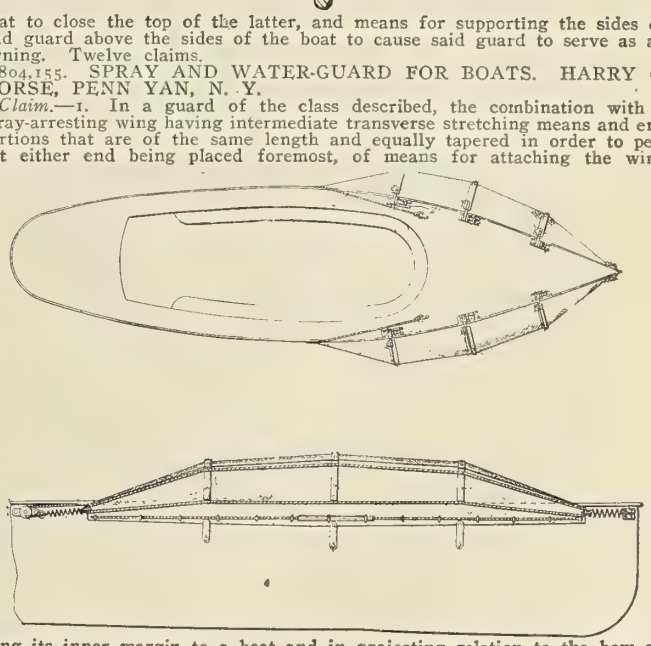
Claim.—1. A boat having a water-guard mounted centrally over its top, means for removably securing the sides of said guard to the sides of the



boat to close the top of the latter, and means for supporting the sides of said guard above the sides of the boat to cause said guard to serve as an awning. Twelve claims.

804,155. SPRAY AND WATER-GUARD FOR BOATS. HARRY C. MORSE, PENN YAN, N. Y.

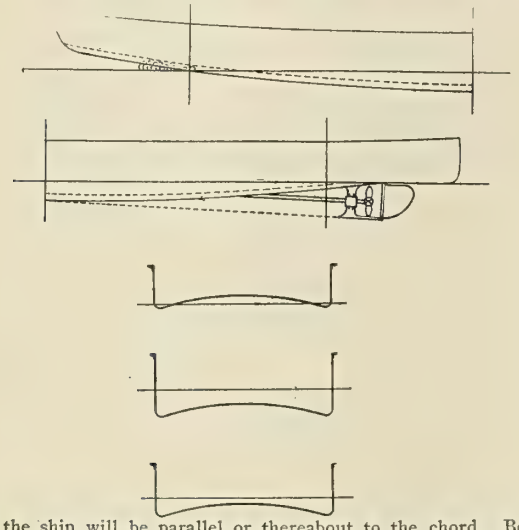
Claim.—1. In a guard of the class described, the combination with a spray-arresting wing having intermediate transverse stretching means and end portions that are of the same length and equally tapered in order to permit either end being placed foremost, of means for attaching the wing



along its inner margin to a boat and in projecting relation to the bow of said boat, and means for securing the tapered ends under tension to the boat. Thirty-seven claims.

804,269. CONSTRUCTION OF THE HULLS OF SHIPS AND BOATS. HERBERT E. ROSS, SYDNEY, NEW SOUTH WALES.

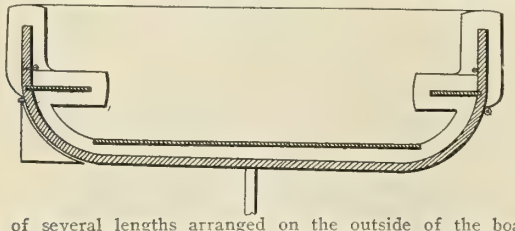
Abstract. The special object of the present invention is to provide such a construction of hull that the vessel shall have a tendency to skim or glide, as nearly as possible, over the surface of the water. To this end the plan of the hull will be an oblong or parallelogram in contour, with vertical or approximately vertical sides depending from the long edges of the parallelogram. The bottom edges of the sides and the external bottom will each form a convex curve, deepest at the center and tapering to zero at both ends, or, briefly stated, each will be a segment of a circle of which the



deck of the ship will be parallel or thereabout to the chord. Between the segmental sides the external bottom will be transversely concaved. The stern may be made flat, so as to lie on the surface of the water, a construction which is at present in use. When screw propulsion is employed, the propeller-shaft may be inclosed and the stern-clearance be of solid deadwood or the shaft may be left bare. One claim.

804,797. DEVICE FOR RENDERING SMALL BOATS UNSINKABLE. JOSEPH M. A. DEYDIER, NYONS, FRANCE.

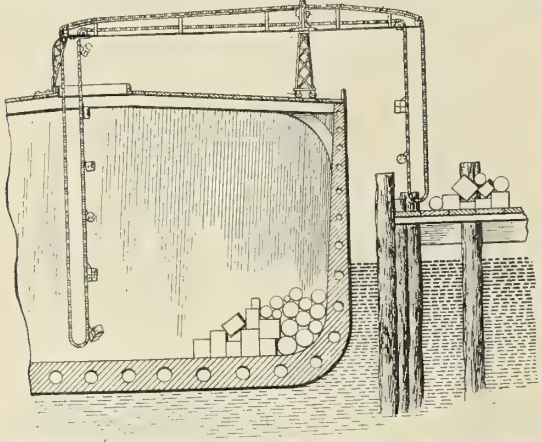
Claim. In an unsinkable boat the combination with the shell of rods



composed of several lengths arranged on the outside of the boat, stuffed waterproof cushions provided with rings intended to be engaged by the said outer rods, means for securing said cushions to the inside of the boat after they have been bent over the edges of the boat-shell, cushions secured to all parts of the inside of the boat and a waterproof covering protecting the cushions placed at the bottom. One claim.

805,711. APPARATUS FOR LOADING AND UNLOADING SHIPS. FRANKLIN B. CLARK, WASHINGTON, D. C., ASSIGNOR TO THE AMERICAN LOADING AND UNLOADING MACHINE CO., (INC.,) OF VIRGINIA.

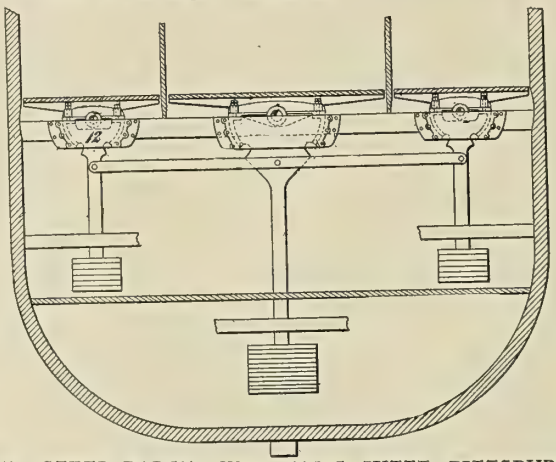
Claim.—2. A loading and unloading apparatus comprising a standard, a truss frame made up of an upper and lower set of tracks connected together and mounted upon said standards, the upper set of said tracks extending beyond the ends of the lower set and downwardly inclined, sprocket wheels



mounted in suitable bearings at the ends of said tracks, an endless carrier mounted to travel upon said tracks and over said sprocket wheels, and a support for corresponding ends of said tracks and through which support said carrier is adapted to travel. Eight claims.

805,718. BALANCED FLOOR FOR SHIPS' CABINS. ROBERT P. DEWEY, SAN FRANCISCO, CAL.

Claim.—1. In a ship, the combination of a movable floor or deck section, a shaft to which the floor section is fixed, a weighted pendulum detached from the floor so as to be independent thereof, and a cross-head or bearing on the shaft and extending beyond opposite sides thereof, said pendulum engaging the cross-head to effect the oscillation of the shaft and floor section and thereby maintain the floor section in horizontal position irrespective of the movements of the ship. Four claims.



805,468. STEEL BARGE. WILLIAM C. JUTTE, PITTSBURG, PA.

Claim.—11. In a barge or other boat, the combination of a side top member comprising an angle-shaped beam having an inwardly-projecting flange at its top edge and having the lower portion of its web offset inwardly, a corner member having the end portions of its top and outer faces offset inwardly to receive the end of said side top member and having the lower edge of its outer face offset inwardly on a line with the inwardly-offset portions of the side top member, and plates riveted to the offset portions of the side top member and at the lower edge of the corner member. Twenty-five claims.

805,496. BUOYANCY-REGULATING APPARATUS FOR SUBMARINE BOATS. LAWRENCE Y. SPEAR, GREENPORT, N. Y., ASSIGNOR TO ELECTRIC BOAT COMPANY, NEW YORK.

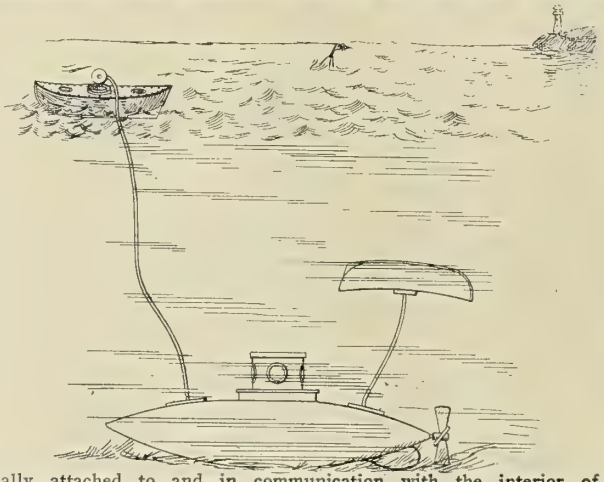
Claim.—1. A submarine boat having means for regulating its buoyancy, comprising a ballast tank, a strong intermediate tank of relatively small capacity, a valve-controlled pipe connecting said tanks, a valve-controlled pipe connecting the stronger intermediate tank with the sea, and means for forcing water from the ballast tank to said intermediate tank and from said intermediate tank to the sea. Four claims.

805,950. GOVERNOR FOR MARINE ENGINES. RODOLPHE J. W. GRASSET, SOUTH YARRA, VICTORIA, AUSTRALIA.

Claim.—2. In an electrically-controlled hydropneumatic governor for marine engines, the combination with the steam valve thereof, of a solenoid having its cores connected directly with said valve and means for energizing the solenoid, said means consisting of an air chamber in communication with the sea, a diaphragm in the upper part of said chamber, a sliding rod connected with said diaphragm, a spring controlling the upward movement of said diaphragm, means for regulating the pressure of said spring, contacts carried by said rod, stationary contacts with which the beforementioned contacts are adapted to engage and wires connecting the stationary contacts with the solenoid. Two claims.

805,975. LIFE-SAVING BOAT. JESSE A. MARLEY, NEW YORK.

Abstract.—This invention relates to life-boats designed especially for use in connection with submarine vessels, and has for its objects to produce a comparatively simple inexpensive device of this character which will be



normally attached to and in communication with the interior of the vessel, and one whereby in the event of the vessel failing to rise after being submerged persons thereon may escape from the vessel and be saved through the medium of the life-boat. Three claims.

On page 25 of the January issue are a number of equations for determining the mean effective pressure in a cylinder under various conditions. One of our readers has pointed out to us that there is an error in the placing of the decimal point in two of those equations, they being respectively the second equation in the first column and the first equation in the second column of that page. In these two cases, the figures given in the numerators of the fractions within the parentheses should be 2.379 and 2.639, instead of .2379 and .2639, respectively.